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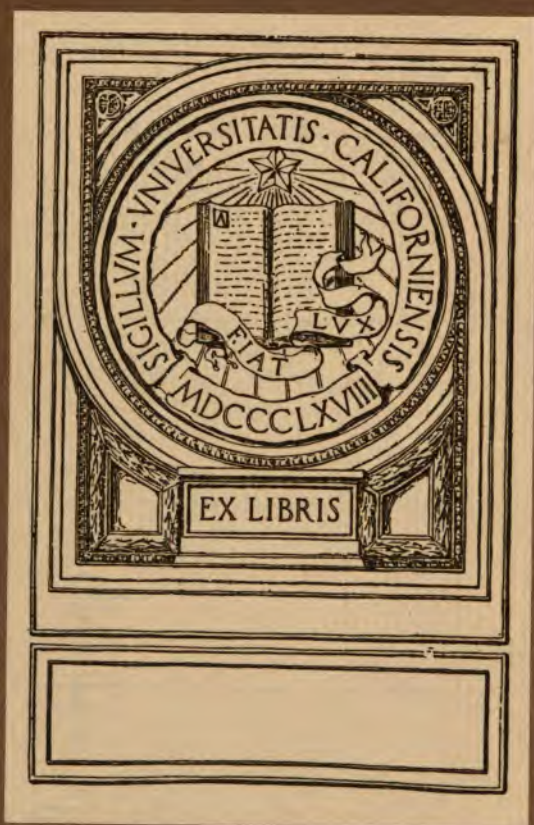
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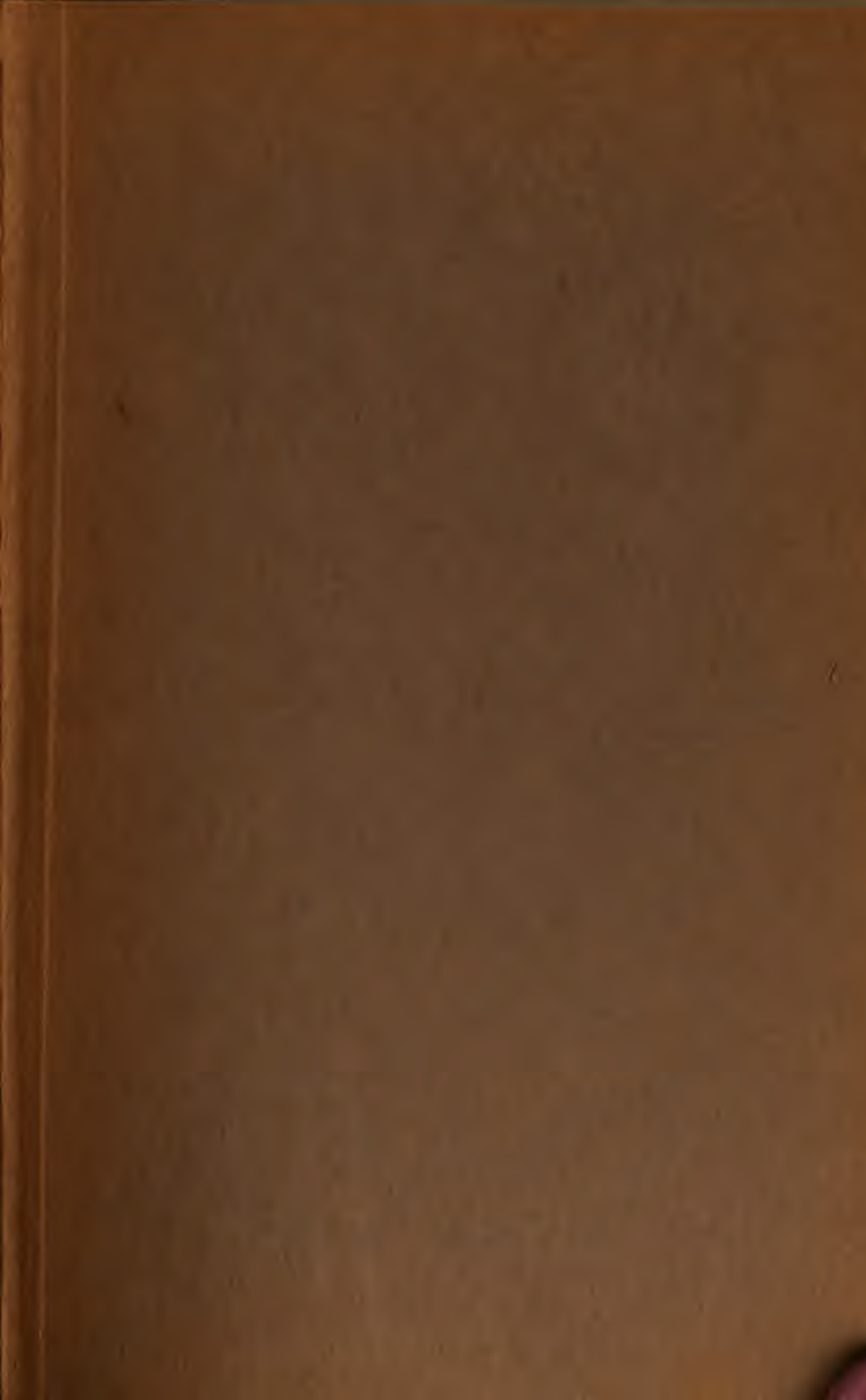
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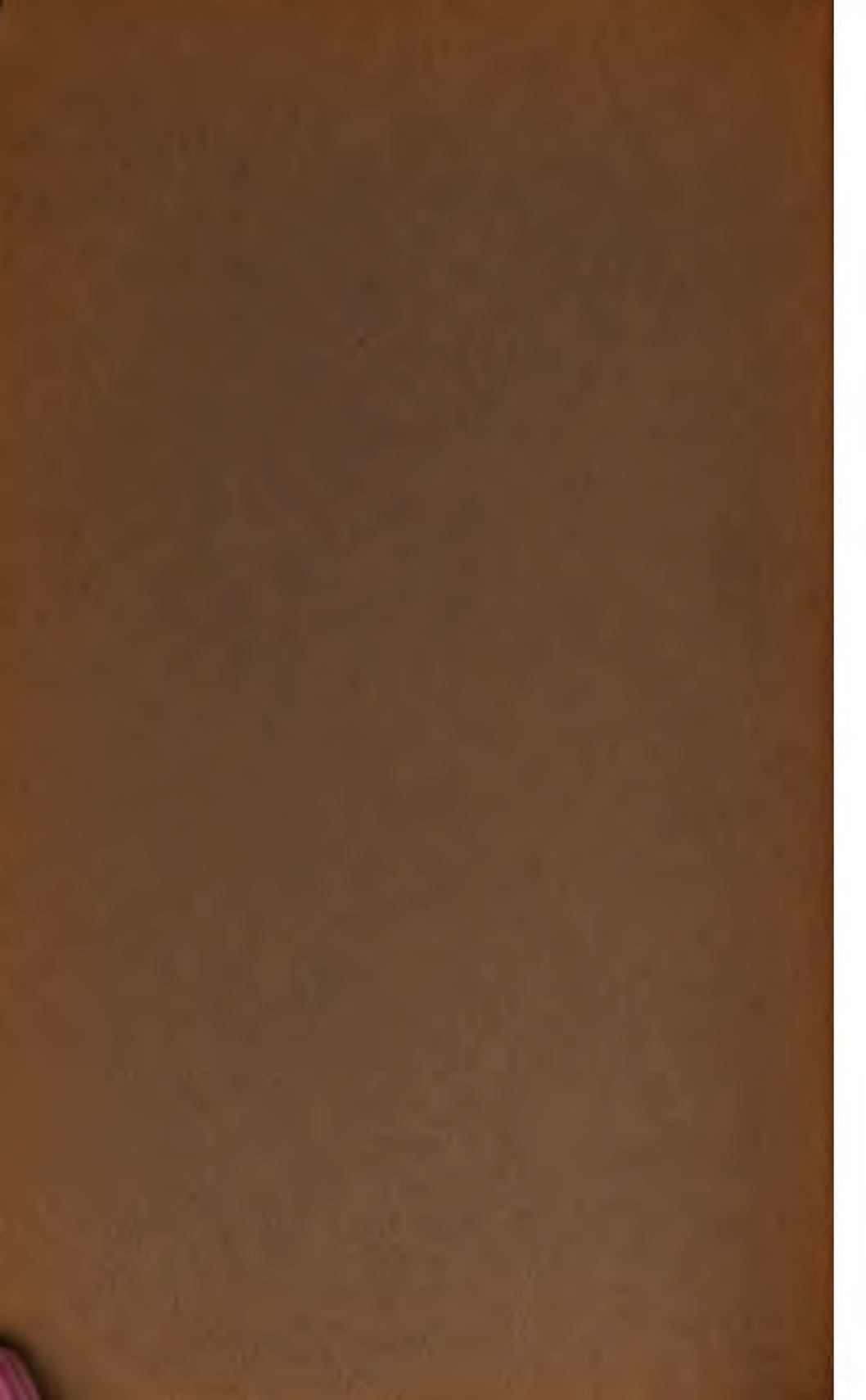
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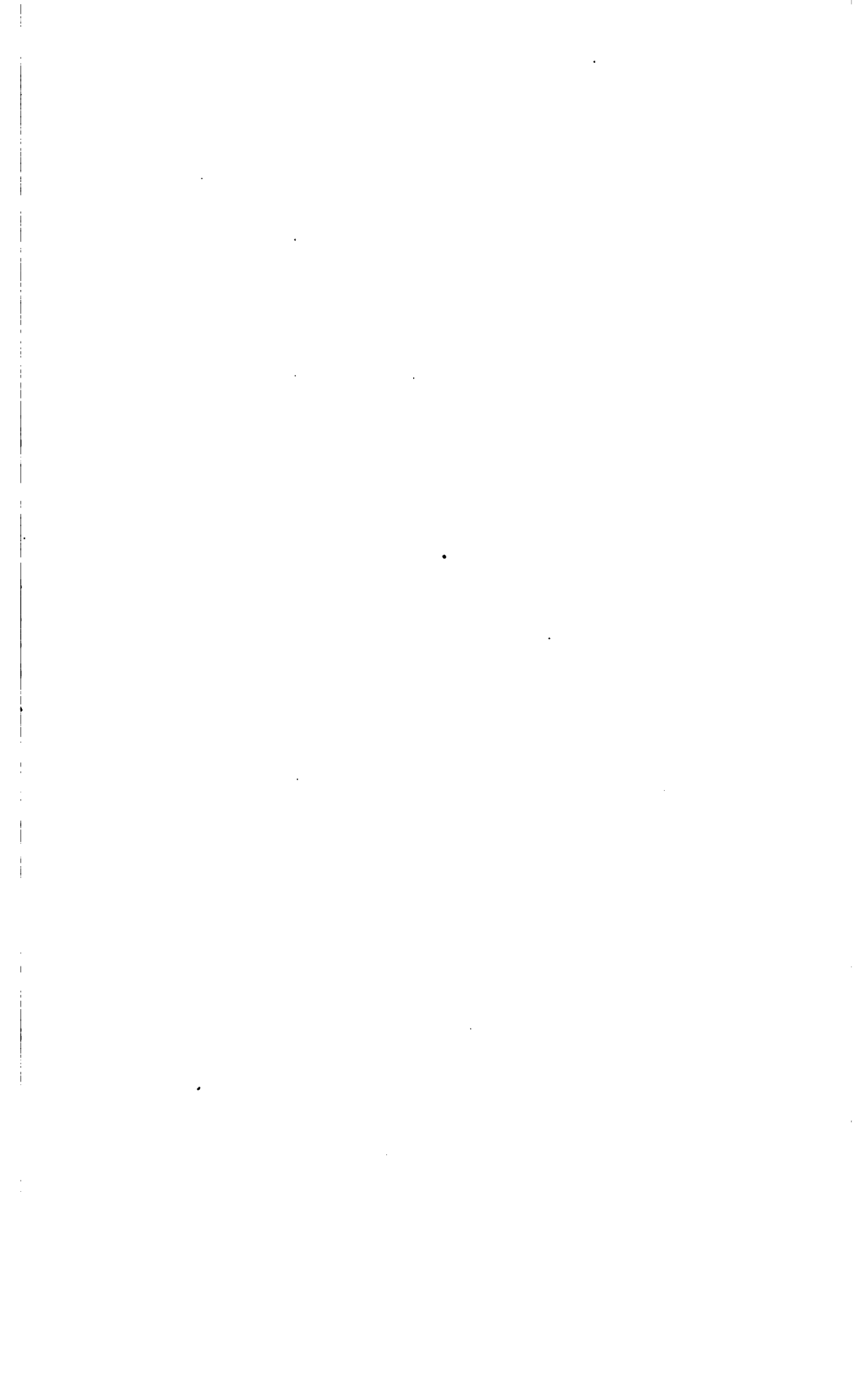
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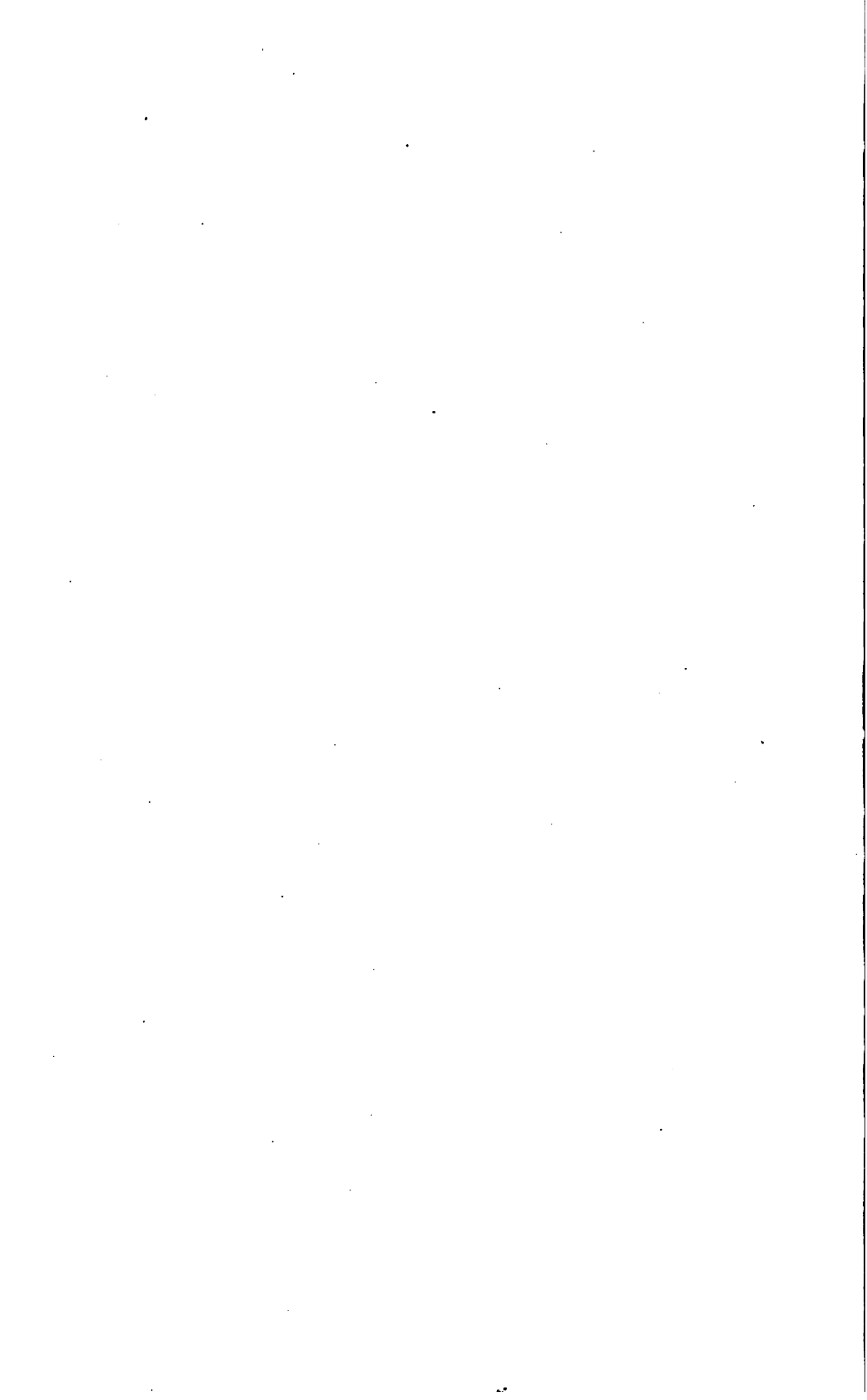
A COURSE OF INSTRUCTION
IN
•ELEMENTARY
MACHINE DESIGN.

ARRANGED FOR USE
IN
TECHNICAL SCHOOLS

BY
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Professor of Engineering Design



PURDUE UNIVERSITY
LAFAYETTE, IND.
1909



PREFACE.

In submitting this book on Elementary Machine Design, the author wishes to state that it represents the course of instruction upon the subject as given to the Junior Engineering students at Purdue University. The material here presented is the outgrowth of a course of lectures, which required much valuable time to deliver and brought only indifferent results. It was considered wise, therefore, to present the notes in printed form and include other problems and designs so as to permit the book to be used as an elementary text. The outline given below has been followed for the last four years with a great deal of satisfaction to students and instructors alike and has proven a great success in the work. The first and second editions were less comprehensive in their scope and were designed for local use only. These editions found a considerable demand from other sources, and it was found to be worth while to revise and enlarge and present it for general use as well. The aim has been to make it strictly elementary, and to add references to the various chapters for more complete study when this is found necessary. There are many treatises on machine design, but these are reference books, and however complete in their development, are not well adapted to aid in teaching the subject of machine design to a class of men, who do not know the first principles of design. Hence, to economize the limited time usually allowed for this subject, it is thought best first to present a simple object lesson so the student will see how such work is carried on, and then assign some subject which is fairly similar, but which will require independent work on his part.

In the preparation for such a course, it is assumed that the student has had a training in mechanical drawing and has at least a beginning in the perception of mechanical symmetry. He should be familiar with shop methods and machines so as to eliminate expensive processes in shop production. He should also be able to apply the principles of mechanism and mechanics to the solution of constructive problems. These things are very important. Drawing and shop work should be considered pre-requisites, while mechanism and mechanics may be taken in parallel with the design. The designs comprehend theoretical analysis in every part where it is possible to apply it. Sketching from models, adapting standard sizes from tables to machine parts, and copying drawings already made constitute, it is believed, more nearly the work of the draftsman than of the designer; the work of the latter being more properly the developing of new and original ideas. After the first two general designs, it will be found that the suggestions for any solution are either very limited

or entirely absent. In this work a machine is studied as a whole rather than by parts. By so doing the relation of any part to its neighbor may be more thoroughly analyzed and its design be more peculiarly adapted to its particular work.

In assigning the work it is not probable that any one machine can be selected which will embody all the features desired in such a course, hence the following outline may be found of value:—

(a) Some form of Toggle Joint or lever press containing members in tension, compression, shear and flexure; represented by Design No. 1 and alternates. Time, including one hour test or lecture each week.....66 hours.

(b) Elementary sheet in Kinematics, represented by some form of cam or link mechanism for a typical machine. This is to give relaxation from the previous assignment. Time, including one hour test or lecture each week.....10 hours.

(c) Some form of power machine which will review the simple stresses mentioned in (a) and include those incident to revolving shafts, gears, etc. See Design No. 2, and alternates. Time, including one hour test or lecture each week..84 hours.

(d) Sheet in Advanced Kinematics. Original problem, link motion or governor diagram. Time, including one hour test or lecture each week12 hours.

(e) Some form of machine using steam, air or hydraulic cylinder; see Design No. 3 and alternates. Time, including one hour test or lecture each week.....20 hours.

(f) Sheet in Advanced Kinematics. Original problem. Time, including one hour test or lecture each week.....24 hours.

(f) Should be optional and may be dropped so as to give more time to (e) if assignment demands it.

The above items will fill very nicely, one year's work, 36 weeks at 6 hours per week.

The author wishes to acknowledge kindly assistance from all his associates in the Department of Design, and to mention especially the valuable suggestions given by Prof. C. B. Veal and Mr. B. F. Raber, who aided quite materially in the arrangement and final proofing. Acknowledgement is also given for the many references taken from the standard works in Machine Design, and to the manufacturing firms that have so courteously responded with information, to all of which the author is greatly indebted.

February 1, 1909.

J. D. H.



Elementary Machine Design

CHAPTER I.

Materials Used in Machine Construction.

One of the first qualifications that the machine designer must have, is a thorough knowledge of the qualities of the materials ordinarily used in machine construction. It is therefore considered necessary that a brief discussion of these materials be given.

1. **Iron Ore**:—The basis in all the iron and steel production is the iron ore. The various ores are: Magnetite, Red Hematite, Brown Hematite, Oxides and Siderite-Ferrous Carbonate. These ores vary in color from black or red to yellowish brown and have theoretical percentages of pure iron of, from 72.4 to 48.3 respectively. Of the ores mentioned Red Hematite is of the most importance in the production of pig iron. It is found in abundance and contains about 70 per cent of pure iron.

2. **Pig Iron**:—Pig iron is obtained by a reduction of the iron ore in the blast furnace. The charge of the furnace is made up of fuel, ore and limestone or flux. The limestone forms a fusible slag with the silica and clay of the ore and is floated off as a scum, while the free iron is taken off through the bottom of the furnace and carried in channels to open molds made in the sand on the floor. The metal when run into the small mold, is called a *pig*, and that in the channel a *sow*.

Pig iron contains a number of impurities, the principal ones being carbon, silicon, sulphur, phosphorus and manganese.

3. **Carbon** in iron is found in chemical combination and also in the free state as graphite. The percentage of carbon seldom exceeds 4 or 5 except in special combinations rich in manganese and chromium, where it is said to be found as high as 7. Combined carbon increases the tensile strength and the hardness of cast iron but diminishes its ductility. The amount of free carbon has no effect on the quality of the iron, but it indicates a metal having more ductility and less tensile strength than where the free carbon is less. The manipulation of the iron determines to a large degree in which of the two states the carbon will be; for example, a piece of iron that has cooled slowly from a melted state presents considerable free carbon, but the same piece of iron with the same sum total of carbon, when chilled or cooled suddenly, presents little if any free carbon. High silicon irons are usually lower in total carbon.

4. **Silicon** unites chemically with iron and is found in all blast furnace irons. Its presence in the charge has a tendency to reduce the amount of combined carbon in the iron, and to increase the

graphite, thus softening the iron and making it more fluid. Silicon, alone in the iron, increases shrinkage and increases hardness, but by increasing silicon and consequently changing combined to free carbon, as would usually be the case, it reduces shrinkage, and softens the iron. The maximum strength of cast iron is probably obtained with 2 to 3 per cent silicon.

5. **Sulphur** unites chemically with iron and is found in all blast furnace irons. Its presence is favorable to combined carbon. It increases shrinkage and above .1 per cent weakens the casting. Tests have been made that showed iron having small per cents of sulphur to be stronger than iron with little or no sulphur in it.

6. **Phosphorus** in small quantities is found in chemical combination with all blast furnace iron. It is not objectionable in small quantities as it makes the metal more fluid, and decreases shrinkage, but in quantities above 1 per cent it seriously weakens the iron without any corresponding benefit.

7. **Manganese** combines with iron in almost any proportion. When iron containing manganese is remelted, the latter decreases by slagging. Increasing manganese increases combined carbon, increases shrinkage and reduces silicon. It also increases the tensile strength and fluidity making the metal harder and more brittle. Spiegeleisen and ferromanganese, special pig irons used in steel making, contain, the former 10 to 25 and the latter 25 to 90 per cent of manganese.

8. **Classification of the Blast Furnace Product:**—The iron obtained by the blast process is the basis of all the commercial irons and steels. The quality and composition of the iron from the various charges differ somewhat and it becomes necessary to assort the iron to suit the special requirements. The classification is usually as follows: *Bessemer* iron, used in the manufacture of Bessemer steel; *basic* iron, used in the basic process of steel manufacture; *mill iron*, used in the puddling furnace, for the manufacture of wrought iron; *malleable* iron, used in making malleable iron castings; and *charcoal* iron and *foundry* iron, used for general utility castings and foundry work.

This classification of the iron may be accomplished in either one of two ways: by fracture, which is the more common way, or by chemical analysis, which is the more scientific way and which without doubt gives more satisfactory results.

9. **Cast Iron:**—The charcoal and foundry irons, usually called pig irons, are shipped throughout the country for use in the foundries as a basis for all gray iron and malleable castings. Pig iron is melted in a cupola with a certain percentage of scrap cast iron, and then poured into a mold, thus becoming a casting. Gray pig iron makes a casting that is very soft and easily machined, but one having a low tensile strength, hence the pig is usually mixed with other grades of cast iron for commercial use.

The process of remelting cast iron has the effect of burning out the free carbon and increasing the combined carbon and sulphur. The result of this process is an increased hardness and a more finely divided crystalline structure each time it is remelted. This mixing and remelting may be carried on through a number of stages from the *gray* pig iron with its open texture, to the *white* cast iron with its fine granular appearance.

The following summarizes the above statements:

Cast Iron 1½ to 4 per cent Carbon	Gray Cast Iron	Grades	
		A	{ High in free carbon; gray color; low tensile strength; soft; fairly tough; when melted, is very fluid and has moderate shrinkage; used in castings requiring heavy machining and little strength.
		B	{ Medium in free carbon; light gray color; strong; easily machined; used mostly on machinery castings.
	White Cast Iron	C	{ Low in free carbon; white; crystalline; hard, brittle; when melted has moderate fluidity and heavy shrinkage; used on heavy machinery castings, castings subjected to excessive wear, and castings requiring little or no machine work, structural castings, etc.
			{ Very close, white granular appearance; stronger than gray iron but more brittle; carbon in chemical combination; used largely for conversion into wrought iron and steel.

10. *Gray cast iron* is the chief metal in machine construction because of its almost universal adaptability. It is more easily and more cheaply formed into intricate shapes than any other metal; it resists oxidation better than wrought iron or steel and it has a high compressive strength. It has, however, the following disadvantages: low tensile strength; cannot be riveted or welded; very brittle, and very liable to hidden flaws and defects.

Cast iron may be *annealed* by heating to a light red and cooling very gradually. Annealing cast iron reduces its strength and increases its toughness.

11. The *shape* of a casting also affects its *strength*. The molecular

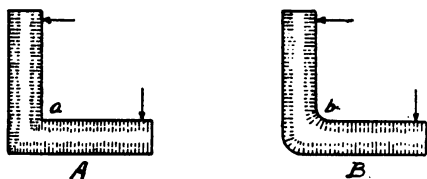


FIG. 1.

formation in a casting takes place when it is cooling in the mold. In certain shaped castings, this formation seems to develop lines of strength and lines of weakness. A simple explanation

of this, although possibly not a correct one, would be the formations of the molecules in lines perpendicular to the surface. If a casting A, Fig. 1 should have forces applied in the direction of the arrows, the angle at *a* would be considered unsatisfactory. At *b*, however, owing to a better arrangement of the lines in cooling, the strength would be decidedly increased.

12. Cast iron *shrinks* about .01 of each linear dimension when cooling in the mold; this quality should be remembered by the designer wherever pattern sizes are specified.

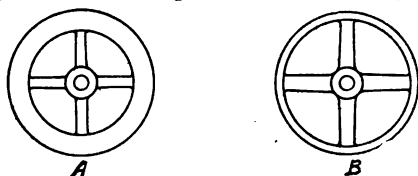


FIG. 2.

13. The designer should so *proportion* a casting that the cooling in the mold will not throw an unnecessary strain upon any part. In A, Fig. 2, the rim of the pulley, being heavy, will set last,

and in shrinking will cause a heavy compression in the arms, with a corresponding tension in the rim. If the arms are weak compared to the rim they may be thrown out of their plane. In B, the arms being heavier than the rim, they will set last and in shrinking will cause a compressive stress in the rim with a corresponding tensional stress in the arms. If the arms are stronger than the rim the latter will be thrown out of a circle, but if the rim is stronger, the arms will crack under any sudden blow; thus relieved, the rim again assumes its circular shape.

14. The *warpage* of a casting may be shown by Fig. 3. The portion *a b*, because of its relative thinness, sets first in a straight line and *c d*, cooling last, will

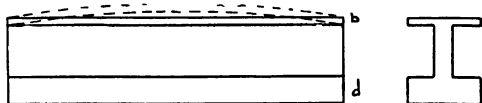


FIG. 3.

draw *a b* away from the original position and cause the face to become convex. Other forms of distortion in castings from the same general causes might be shown but this one is considered sufficient. When it becomes a

necessity to so proportion a casting that there will be evident distortion from warpage, the designer may anticipate this distortion by building up on the parts thrown out, so that the final casting may have the desired shape. The designer should not overlook the fact, however, that there will be a heavy internal stress in the metal whenever warpage exists. The preferred way would be to proportion the casting so the cooling strains would be neutralized.

15. Wrought Iron:—Mill iron is placed in a refining or open hearth furnace and melted under a strong draft. Most of the impurities in the iron are burned out as it melts and it finally becomes a thick spongy mass at the bottom of the furnace. This is collected in the form of a *sponge ball* or *bloom* on the end of an iron bar and is taken to a squeezer where the slag or dirt is squeezed out of it. After this the bloom is subjected to repeated heatings, hammerings and weldings, until it is of a uniform texture, when it is finally drawn out under hammer or roll into the various shapes of the commercial bar iron.

Pure wrought iron contains no carbon. It is seldom produced in a pure state because of its extreme softness which renders it of little practical value in commercial work. Carbon is the hardening agent, and in some grades of wrought iron it may be found as high as .2 per cent. Where carbon is found in excess of this value the metal is classified as a mild steel.

The *quality* of the wrought iron depends upon, first, the removal of the carbon, and second, the care with which the bloom is worked up. The latter determines the grain or fibre of the metal. The more thoroughly the bloom is worked up the more homogenous will be the metal and the better the quality. Rolling and hammering iron increases its strength because it increases its density, hence the ultimate strength of wires and small bars is greater than that of larger bars. When wrought iron is rolled cold under heavy pressure it obtains a smooth polished surface and is greatly increased in strength.

Wrought iron is tough and elongates about .0001 of its length for each ton of load per square inch of section up to the limit of elasticity; beyond this point it elongates much more, the greater part of the elongation being permanent. When a bar is broken under tension it usually draws out at the point of fracture. This reduction of area is sometimes used as a measure of the fitness of the material to perform certain duties. In comparison with cast iron it is more satisfactory under tensile and transverse stress, but less satisfactory under compressional stress. It cannot be melted and run into molds like cast iron but it can be welded and forged into any desired shape.

Wrought iron cannot be tempered like steel but it may be *case hardened* by heating to a white heat and covering the surface with cyanide of potassium or ferro-cyanide of potassium and dipping

in water. The cyanide slightly carbonizes the surface of the metal and gives it a quality about the same as steel. The chilling of the steely surface by the water, then, actually tempers it.

16. Malleable Cast Iron:—Malleable castings are obtained in two ways. The first and oldest method is by making a cast iron casting from white or mottled charcoal iron in an ordinary mold, and then subjecting this casting to a high heat in the presence of iron oxide. The carbon in the surface of the casting combines with the oxide of iron which surrounds it and leaves the surface of the metal with much the same qualities as wrought iron. In the process of decarbonization the castings are packed in cast iron or wrought iron boxes with red hematite or peroxide of manganese, and piled in a furnace; they are then kept at a bright red heat for several days after which they are allowed to cool slowly.

The second method is by the open hearth process. Malleable pig iron is melted in the open hearth furnace under blast. This process of decarbonization is carried on until the metal is of about the quality of semi-steel, when it is run into molds. Castings made in this way are of a more uniform texture and are stronger than those made by the first process.

Malleable castings are soft, tough, strong, flexible, and can be easily machined. Good specimens may be bent double, but could scarcely be bent back again.

To make a good malleable casting by the first process the shape should be such as to permit the greatest amount of surface in contact with the iron oxide. A thin rectangular, star, or ribbed section is better than a square or round section. The effect of decarbonizing is to produce a soft external malleable shell around a hard brittle core of cast iron. The thinner the section, the greater will be the percentage of malleable iron as compared to the cast iron, and the more tough and flexible will be the casting.

Malleable castings find their greatest usefulness in agricultural machinery, wind mills, small machine parts, ornamental work, and car and locomotive work.

17. Steel:—Steel is a chemical combination of iron and carbon and may be made by any one of three processes: the *crucible*, the *Bessemer* and the *open hearth* process.

18. Crucible Steel, or cast steel is produced by remelting blister steel (pure wrought iron bars heated in contact with charcoal or carbon until it absorbs a certain percentage of carbon) in a crucible and then pouring into molds. It is also produced by melting pure wrought iron in a crucible with enough charcoal and cast iron to introduce the required amount of carbon.

This grade of steel contains from .4 to 1.5 per cent of carbon, .1 to .75 per cent of manganese, .1 to .2 per cent of silicon, .01 to .02 per cent each of sulphur and phosphorus, and is used in mak-

ing springs, cutlery, machine tools, and such machine parts as require hardening. The best all around tool steel has about 1 per cent carbon. Razors, lathe tools, drills, etc., have 1 to 1.5 per cent.

19. *Bessemer Steel* is produced in a Bessemer Converter by forcing a powerful blast of air through melted Bessemer pig. When most of the carbon is burned out small quantities of cast iron and spiegel containing a known amount of carbon are added to bring the carbon of the mixture up to a definite amount. This when thoroughly mixed is poured into ingot molds and becomes the basis for some of the cheaper grades of steel, such as, rails, nails, light shafting, merchant bar and some grades of rolled plate.

20. *Open Hearth* or *Siemens-Martin* steel is produced by melting a certain amount of basic pig iron with wrought iron or Bessemer steel scrap. The composition and manufacture vary somewhat according to the available scrap, the latter being replaced occasionally by iron ore, spiegeleisen or ferro-manganese. Open hearth steel is used in structural work, forgings, car axles and the better grades of steel plate.

21. All soft or mild steels are either Bessemer or Open Hearth. The former is produced more cheaply but the quality is sometimes inferior. On account of the cheapness of both these steels they have replaced wrought iron in most commercial work.

Mild steels generally contain low percentages of carbon, say from .1 to .6 per cent, and approach wrought iron in workability.

The *Quality* of steel is generally considered from the standpoint of the amount of carbon in it. Mild steel is soft, flexible, easily forged or machined and cannot be tempered. Cast steel, or steel high in carbon, is hard, rigid, not easily forged or machined and is readily tempered by heating and suddenly quenching in water. Exceptions to this are some of the self-hardening steels, as Mushet's steel or Hadfield's manganese steel, which harden by heating and cooling slowly in the air. All steels have a higher tensile and compressive strength than wrought iron or cast iron.

22. Mild steel may be somewhat hardened by hammering and rolling, as would be true of any soft metal, like wrought iron. This is due to the increased density of the surface material from the hammering or the rolling.

Steel is *annealed* by heating to a low red heat and cooling very gradually in a bath of dust, ashes or lime. High carbon steel should never be heated above a *low red heat*.

The *fracture* of steel presents a surface that is white, crystalline and fine grained. It has no fibre and usually gives a flat break.

23. **Steel Castings:**—Steel castings take the place of cast iron castings where great strength is required. The quality of the steel is varied to suit the requirements. Castings having about .1 per

cent carbon are soft, tough and ductile; while those having .75 per cent are very strong and rigid. Ordinary steel castings contain from .2 to .5 per cent carbon. *Silicon* varies with the carbon in steel castings, from .1 to .4 per cent; *Manganese* from .5 to 1 per cent; *Phosphorus* .5; and *Sulphur* from .025 to .05 per cent.

Steel castings are most often produced by the open hearth process although they may be made by both the Bessemer and the crucible process. They are used for such works as gears, hydraulic cylinders, engine frames and parts, locomotive driving wheel centers, large rolls, rolling mill spindles, coupling boxes, hammer heads and dies, and ship and railroad castings.

Steel castings are more difficult to make than iron castings because of the unusually heavy shrinkage. They are frequently honeycombed and lack homogeneity, consequently a steel casting is usually made much larger than the size of the finished product to allow for sufficient machining. Castings high in carbon are less liable to be honeycombed than low carbon castings.

24. Alloys of Steel:—Some of the principal alloys of steel are Manganese, Nickel, Chromium, Tungsten, Aluminum and Vanadium.

25. Manganese Steel—Manganese in steel has little effect below 1.5 per cent, but from 1.5 to 7 per cent the strength and ductility decrease and the hardness increases. At 7 per cent it is very brittle. From 7 to 14 per cent the strength and ductility again increase. The ductility is also brought out by sudden cooling. The best results with this steel are obtained at about 14 per cent. Carbon is present in about 1 per cent. Manganese steel is free from blow holes, welds with difficulty, cannot be softened, and combines extreme hardness and toughness. The latter quality is one not found in any other metal and makes this steel very valuable for certain kinds of work. It is used for pins in elevator buckets, chain elevator links, jaws and plates on crushing machinery, safes, car wheels, axles, etc. The ultimate tensile strength of the best manganese steel is about 140,000 pounds per square inch.

26. Nickel Steel is made by the open hearth process and is used principally for armor plates. It is also very often used in commercial forgings and castings requiring very great strength and ductility. Nickel combines with steel having .25 to .4 per cent carbon, between the limits of 2.5 and 4 per cent. Nickel steel tubes are made containing 30 per cent nickel. It does not crack readily, has a high elastic limit, is non-corrodible, is only slightly magnetic and seems to combine the toughness of raw hide with the strength of steel. Nickel steel is about 50 per cent stronger than ordinary steel having the same percentage of carbon.

27. Chrome Steel.—When 2 to 4 per cent of chromium is added to steel containing 1 to 2 per cent carbon, the steel becomes

very hard and is able to resist severe shocks. Chromium is often added to nickel steel, making nickel-chromium-steel. Its principal use is in the production of armor plate and projectiles.

28. Tungsten Steel or Mushet Steel is a self-hardening steel, i. e. will harden by heating to a red heat and cooling in air. This is one of the important steel alloys. The chemical compositions according to Mr. F. Reiser in "Stahl and Eisen," January 15, 1903, are, in per cents, about as follows: tungsten 5, manganese 2.5, carbon 2, chromium 0.5, and silicon 1.3.

Tungsten Steel is very hard and when hot becomes very brittle. It can only be worked between an orange and a bright orange heat, and then with extreme care. It is used chiefly for machine tools and since it is so hard that it cannot be worked cold, it is generally produced at the mill in standard sizes which require only grinding before using.

Some tests of tungsten steel by Styffe showed an ultimate tensile strength of 172,000 pounds per square inch.

29. Aluminum Steel.—Aluminum combines with steel in any proportion up to 15 per cent. It has no very important action upon the mechanical properties of steels when the percentage is low. Above 2 or 3 per cent it causes brittleness in the metal.

30. Molybdenum Steel.—The principal use of this steel is in tools. It is also used in large cranks, propeller shafts, gun barrels, boiler plates and wires. Molybdenum increases the elongation of steel very greatly. This is of special importance in the production of wires. It is usually found in combination with the nickel steels.

31. Vanadium Steel.—0.1 to 0.2 per cent vanadium raises the elastic limit and tensile strength of mild steel 25 per cent or more. It appears to retard segregation and thus enables the steel to resist deterioration under continued vibration. Vanadium steel is being increasingly used for axles, piston rods, crank shafts and other transmission machinery.

32. High Speed Steels.—The exact chemical compositions of high speed steels are unknown, except to the makers. According to Prof. L. P. Breckenridge in Bulletin No. 2, University of Illinois, on "Tests of High Speed Tool Steels on Cast Iron," the following elements are found in varying quantities: carbon, tungsten, chromium, manganese, molybdenum and titanium; the Taylor-White steel having as high as 12 per cent tungsten and 4 per cent chromium.

When these steels were first produced, they were supplied to the trade in an unannealed state, but now they are usually annealed. The advantage of the latter over the former is that it hardens better and is less liable to fail from internal stresses set up by the hammering and rolling process. Annealed stock may be nicked and broken, or it may be forged in an ordinary forge fire between a bright red and a bright yellow heat. It should never be hammered below a

bright red heat. To harden this steel, it should be heated until it approaches a clear white heat, or at least until the flux begins to run on its surface, and then it should be cooled in an air blast, or quenched in a bath of lead or oil. The latter is in many cases preferred since it makes a tool that is tougher and one that will resist greater shocks.

In the use of high speed steels a great increase in the work output can not be expected so long as the steel is used in the old design of machines. It is possible to increase the cutting speeds of metals as high as 300 or 400 per cent above that of the carbon steel, and as high as 100 per cent above the so-called self-hardening steel. It is possible also, to multiply the size of the cut as many times. Under such conditions it is apparent that machines designed for carbon or self hardening steels will need redesigning before they can be successfully forced to the full capacity of the high speed tool.

33. Copper:—Pure copper is red in color, is ductile and malleable and can be forged hot or cold. It is cast in blocks and is then drawn out under hammers or rolls. Hammering and rolling increases its strength and brittleness. The latter quality may be removed, however, by annealing. Copper is annealed by heating and suddenly cooling. It may be hardened by heating and cooling slowly. In these particulars it is the opposite of the other metals. Hard drawn wire has about three times the tensile strength of cast copper, and is nearly equal to that of mild steel. When hard drawn copper is annealed it loses about 25 per cent of its strength.

It is difficult to get a sound copper casting. Good copper castings may be made somewhat more readily by the addition of 1 to 3 per cent phosphorus.

34. Copper Alloys:—There are a great number of copper alloys. These alloys vary slightly in composition, but are known under the general name of *bronzes* or *brasses*. Of the bronzes there are compositions known as *Phosphor bronze*, *Aluminum bronze*, *Silicon bronze* and *Manganese bronze*. Of the brasses there are those known as, *high brass* and *low brass*.

Formulas representing each of the above are as follows:

Phosphor bronze, Copper 89%, Phosphorus 1%, Tin 10%.

Aluminum bronze, Copper 90%, Aluminum 10%.

Silicon bronze, Copper 96%, Silicon 4%.

Manganese bronze, Copper 67½%, Manganese 18%, Zinc 13%, Silicon .5% and Aluminum 1%.

The following composition, usually termed bronze but sometimes called *Gun Metal*, is a very common one.

Copper 90%, Tin 10%.

In the brasses, the color usually designates the quality; high brass, yellow; low brass, red. The first shows low copper and high zinc or tin, and the second shows high copper and low zinc or tin.

High brass (ornamental work), Copper 65%, Zinc 33%, Lead 2%.

Low brass (bearings, etc.), Copper 80%, Zinc 10%, Tin 5%, Lead 5%.

35. Babbitt Metal:—Some journal boxes, loose pulleys and other friction surfaces that are not considered of the highest order, are lined with babbitt metal, instead of bronze or brass. This makes a cheap and a fairly substantial bearing. Babbitt metal is the cheapest anti-friction metal known, and always contains two or more of the following ingredients: Tin, copper, zinc, antimony, and lead. It is probable that the original formula for this metal was about 90% tin, 3% copper, 7% antimony.

Copper is not always used, because it renders the bearings more liable to friction. Some of the cheapest grades of babbitt metal have a large percentage of lead. This is usually mixed with tin or zinc. The ordinary grades sold in the market have a composition of about 45% tin, 2% copper, 13% antimony, 40% lead.



CHAPTER II.

Notes and Formulas in Elementary Machine Design.

36. Machine Design:—In taking up the study of the design of machine parts our attention is called to the *Concrete Side* of Mechanism and Mechanics. *Machine Design* is an analysis of the definite forces acting upon pieces of material and the proper proportioning of these pieces for symmetry, strength and rigidity. In all such work graphical, as well as analytical, methods must be employed.

37. Graphic Statics:—Statistical problems are those which deal with the equilibrium of forces acting upon bodies at rest.

In the understanding of the following methods, two general statements concerning forces should be kept in mind; viz.:

- (1.) Any force has a definite effect on a body no matter if that body is in motion or at rest.
- (2.) Any number of forces acting on a body will cause it to remain at rest, or to move in the direction of the resultant of the forces.

In order that a body or a structure shall remain at rest, it is necessary that all the forces acting upon it should balance each other, i. e. that the forces should be in *equilibrium*. This fact is used to aid in the solution of statistical problems. The effect of the combined action of all the *known* forces acting on a body enables the magnitude and direction of the remaining force which holds them in equilibrium to be determined.

A force is determined when its magnitude, direction, and line of action are known, and accordingly it may be graphically represented by the length, direction, and position of a straight line. This graphical method is a very convenient, rapid, and accurate method of combining and resolving the action of forces, and the importance of these advantages causes it to be greatly preferred, as compared with the analytical method, in practical work.

Forces are given in pounds, or may be reduced to some equivalent measure of weight, while the lengths of lines are measured in linear units, for convenience, usually inches. Thus, a force of 200 pounds may be represented as a straight line 2 inches long, in which case the scale is 1 inch to 100 pounds.

The *resultant* of two or more forces is a single force which produces the same effect as the forces themselves, and may therefore replace them. Thus, if the straight lines OA and OB , Fig. 4, represent

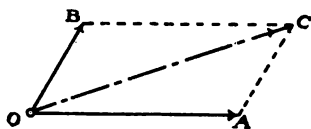


FIG. 4.

two forces acting at the point O , the diagonal OC of the parallelogram $OACB$, called the *parallelogram of forces*, will represent in magnitude and direction the *single force* which will produce the same effect as the two

forces OA and OB acting together, i. e., OC will represent the resultant of the forces OA and OB .

It will be seen that it is not necessary to construct the entire parallelogram, since the triangles on the opposite sides of the diagonal are equal. Either of these triangles can be independently drawn, and when so constructed is called the *force triangle*, or *triangle of forces*.

Often the lines of action of the given forces form part of a diagram upon which it is not desirable to construct the force triangle; in this case, let the three forces AO , BO , and CO , Fig. 5, act at the same

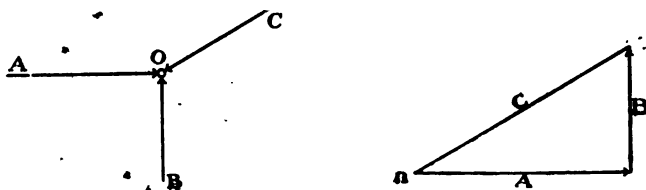


FIG. 5.

point, O ; select any suitable point n for the vertex of a triangle, Fig. 5, in which the three sides A , B , and C , taken in order, represent in magnitude and direction the three forces AO , BO , and CO . Then the three forces will balance one another, that is, they will be in equilibrium; thus the side C of the triangle may be regarded as representing the magnitude and direction of an opposing force, which just balances the forces AO and BO and holds them in equilibrium. If for any reason the triangle does not close, then the forces are not in equilibrium.

To find the resultant of more than two forces we may proceed in a similar manner, the process being simply an extension of the principle of the force triangle. Thus, when it is required to find the resultant of a number of forces acting in the same plane and having a common point of application, the resultant of two of the forces may be found as already explained, a third force may then be united with

it to obtain a second resultant, and this operation continued until all the forces are combined. It is, however, not necessary to construct the various partial resultants in order to find the total resultant. Thus, if we have four forces, AO , BO , CO , and DO , Fig. 6, acting upon the point, O ; these forces, if they are in equilibrium, can be represented in magnitude and direction by the sides of a polygon, whose sides are respectively parallel to the forces. Since a polygon, Fig. 7, so constructed does not close itself, the system of forces is not in equilibrium. To close the polygon draw the dotted line, R , which

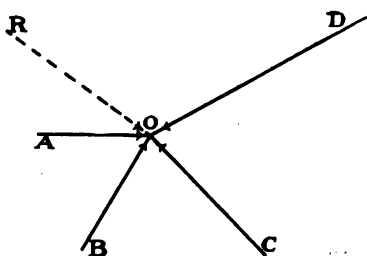


FIG. 6

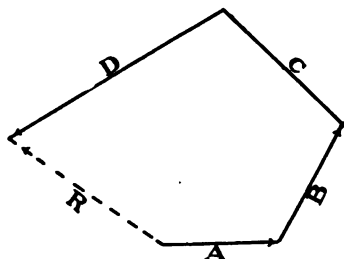


FIG. 7

gives the magnitude and direction of the resultant of the four forces under consideration. The direction of the resultant, R , is opposed to the direction of all the given forces in following around the sides of the *force polygon*; thus the arrow on R has the reverse direction of the other arrows. To produce equilibrium with the original four forces, a force, RO , Fig. 6, equal and opposite to R , Fig. 7, must be applied at O , as indicated. This added force in the force polygon is equal to R with its former direction reversed and closes the polygon of forces.

The foregoing discussion has assumed that the forces under consideration all act at the same point. When several forces lying in the same plane are in equilibrium and have different points of application, so that their lines of action do not intersect at the same point, the force polygon must also close. If the force polygon does not close, then the line joining the initial and final points represents the intensity and direction of the resultant, but its line of action is not determined. This line of action can be found by means of another diagram, called the *equilibrium polygon*.

Suppose several forces represented by P_1 , P_2 , P_3 and P_4 , Fig. 8, be given in magnitude, direction, and line of action; let it be required to fully determine their resultant. First construct the force polygon 1-2-3-4-5, where the length of the closing line 1-5 represents the magnitude of the resultant. Then choose any point, O , called the *pole*, and draw the lines $O1$, $O2$, $O3$, $O4$, and $O5$, some-

times calls *rays*. Now through any point x on the line of action of P_1 , draw a line, xw , parallel to O_2 , intersecting P_2 at w . From w to u draw a line parallel to O_3 ; then draw u to m , parallel to O_4 ; mn , parallel to O_5 , and xn , parallel to O_1 , and the intersection with mn will give the point, n , through which the resultant must pass. If there be applied at the point n a force P_6 equal to R ,

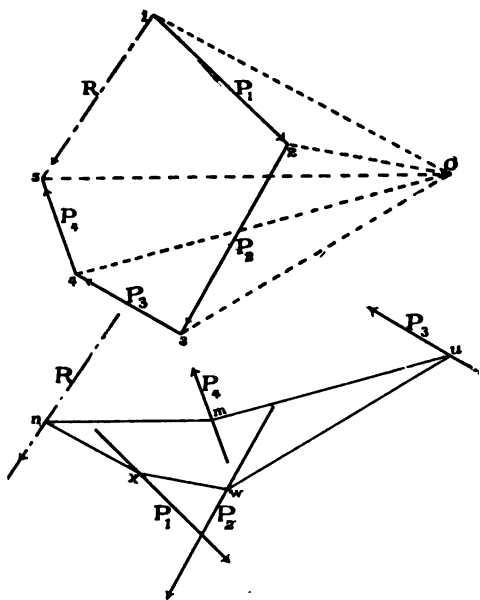


FIG. 8

but opposite in direction, the several forces are in equilibrium and the force polygon closes. The polygonal frame $xwummn$ thus holds the given forces in equilibrium, hence it is called an equilibrium polygon. The graphic condition of equilibrium for several forces not meeting at the same point may be expressed by saying that both the force polygon and the equilibrium polygon must close.

The pole, O , may be taken either within or without the force polygon as may be most convenient for the solution of the problem under consideration, since the position of the pole does not affect the result, as will be found by choosing several poles and observing that the position of the resultant is not affected thereby.

When a number of parallel forces are in equilibrium, the same method may be used, but the diagram becomes simplified, since the force polygon becomes a straight line. Under this condition the equilibrium polygon also has the important special property, that the bending moment in any section parallel to the forces is equal to the

ordinate in the equilibrium polygon at that section, multiplied by the perpendicular distance of the pole from the parallel forces in the force polygon.

Hence by adopting suitable scales the values of the bending moments can easily be found from the diagram.

38. Graphic Statics of Framed Structures:—As the values of forces acting on rigid bodies may be determined graphically, so may the stresses in the various members of framed structures be investigated. Framed structures are of very general application since their use extends from the simple trussed beam or the lever of a machine to the complex construction of the bridge or roof truss.

In determining the stresses in framed structures there are three general methods of applying the laws of equilibrium.

- 1st. To the structure as a whole.
- 2nd. To any single joint.
- 3rd. By passing a section through the structure, removing one portion and applying the laws of equilibrium to the remaining portion, the stresses in the members cut being replaced by equal external forces.

Structure as a whole—To illustrate the use of the equilibrium polygon in determining a portion of the external forces as applied to the structure as a whole, let there be considered the triangular roof-truss, Fig. 9, with dead loads P_1 to P_7 and wind-pressures W_1 , W_2 , W_3 and W_4 .

First, find the abutment reactions with the truss fixed at both ends.

Construct the force polygon, 1-2-3-4-5-6-7-8-9-10-11-12, which shows 1-12 to be the resultant of the given forces. Select any pole O ; draw the rays $O1$, $O2$, $O3$, etc., and construct the equilibrium polygon, npqrstuvwxy, beginning at any point, n , on R_1 , parallel to 1-12. The line OA drawn parallel to mn , the closing line of the equilibrium polygon, gives the required reactions; R_1 equals $A1$, and R_2 equals 12- A .

Second, consider the left end to be on rollers and the right end taking all the horizontal thrust.

Under this condition the vertical component R' , equal $B-1$ is the only part of R_1 acting upon the truss as a reaction at the left end. Its horizontal component must be taken up at the right end support where the truss is fastened to the abutment. Then the resultant reaction R'' , equal 12- B , is made up of this horizontal component AB , combined with R_2 . The reactions in this last case might have been found directly by an application of the force and equilibrium polygons.

Single joints:—To illustrate the graphic method of determining stresses as applied to single joints we will take the following example.

Required a graphical analysis of the Fink roof-truss shown in Fig. 10, the left end of the truss being supported on rollers, the right end fixed.

In this method first find the abutment reactions algebraically, or graphically as explained above, then, commencing at one abutment, find the stresses in the members at successive joints by means of

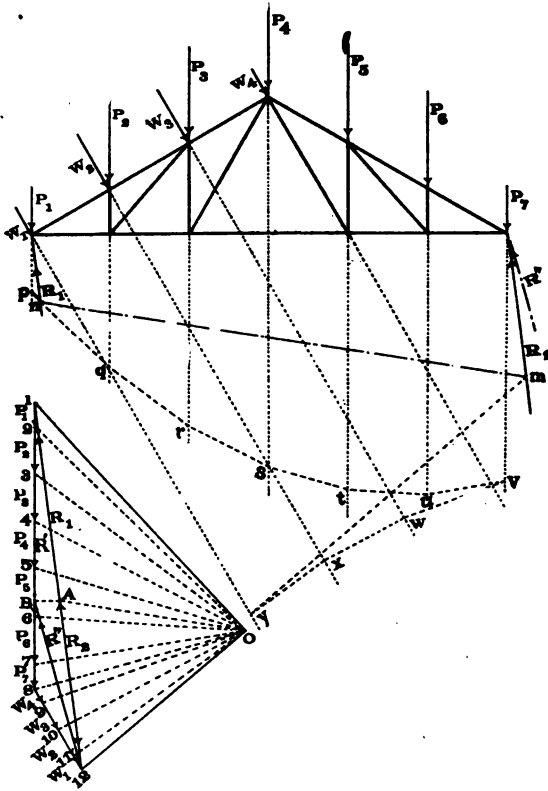


Fig. 9.

force polygons. Instead of drawing a separate figure for each joint, we may combine the force polygons into a single figure, usually called a *stress diagram*. In Fig. 10, each triangle of the truss, and also each space between the external forces is lettered, this being a convenient notation to accompany the method. Each piece and each external force is then known by the two letters in the adjacent spaces. To aid in distinguishing the nature of the stresses, lines representing compressive stresses may be made heavier than those representing tensile stresses.

After drawing the truss to scale, proceed to draw the diagram for *dead load*, due to weight of truss and supported roof, (A) Fig. 10. The "joint-loads" are laid off to form the "load line," aa' . The abutment reactions are $a'm$ and ma . It is necessary to draw but one half the stress diagram where the loads are symmetrical, since the stresses in corresponding members of the two halves of the structure are equal, and the complete stress diagram will be symmetrical with respect to a horizontal axis through m , as shown by the figure. For this very reason, however, it is best to draw the entire diagram where accurate results are desired, in order that the symmetry of the figure may act as a check on the work.

The diagram for *snow load* will be a figure similar to the one for dead load, and the stresses in the two cases will be proportional to the corresponding loads. Therefore, if we multiply each dead-load stress by the ratio between any snow apex load and a corresponding dead apex load, we will have the corresponding snow load stress.

In determining the *stresses due to wind* when one end of the truss is fixed and the other free it is necessary to construct two diagrams, one for the wind blowing on the fixed side and the other for the wind load on the free side, since the abutment reaction at the roller end must in both cases be vertical, and the stresses produced in the two cases will therefore not be symmetrical. The diagram for wind from the left is shown in (B) Fig. 10, where $a-e'a'$ represents the load line, and ma the abutment reaction at the free end. The value of this reaction is most easily found by computation, *i. e.*, taking moments about the left support and considering the total wind load concentrated at the middle of the rafter; the force polygon $a-e'a'-m$ then gives the other reaction, $a'e'-m$; or, since the two reactions and the resultant of the wind loads must meet in a point if produced, (in order that the truss be in equilibrium under the action of these three external forces), we have an easy graphic method of determining the reactions, providing the point of intersection of the three forces comes within the limits of the drawing board, for the direction and line of action of one reaction and a point in the line of action of the other reaction are known. Beginning at a the stress diagram is readily constructed.

(C), Fig. 10, is the diagram for wind from the right. Here $a'-ea$ represents the load line. The reactions may be obtained analytically, or graphically, and the diagram then drawn.

Unless special care is exercised in drawing the wind stress diagrams, they will not close. As they lack the check of symmetry it is not so easy to locate the error, and therefore it is best to construct new diagrams until one is obtained that closes properly.

The maximum stress of each kind in each piece may now be obtained by combining with the dead-load stress, whatever possible

combination of the snow and wind load stresses will give the greatest total tension and the greatest total compression.

Free section:—No special problem need be taken to illustrate the third method indicated above, *i. e.*, applying the graphical analysis to a part of the structure, using successive sections commencing at one end.

If we pass a section through a structure, cutting but two members whose stresses are unknown, the single condition that the force polygon, drawn for the forces acting upon one portion of the struc-

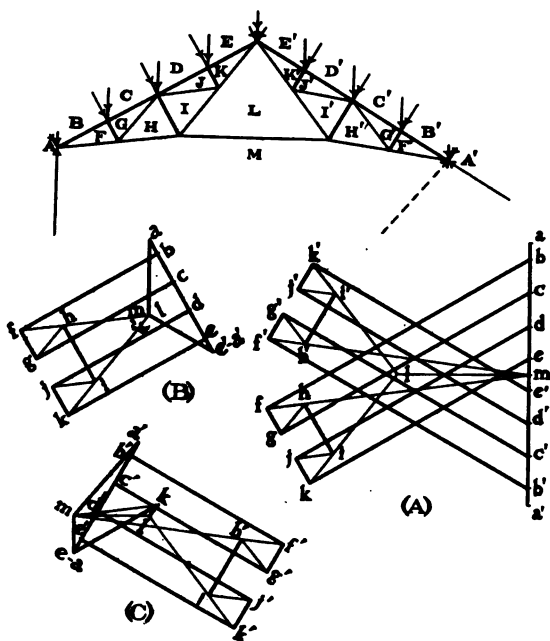


FIG. 10

ture, must close, will enable us to find the stresses in these pieces. Commencing at one end of a structure and passing a section cutting but two pieces, one can determine their stresses; then passing another section cutting three members, one of which has already been treated, the stresses in the other two may be found and finally by successive sections all the stresses may be determined by simple force polygons. While this is different than that given as the second method, the resulting diagram is precisely the same as would be obtained by that method; moreover, by passing any section whatever through the structure, the polygon of the forces acting upon either portion will be given by the diagram.

To investigate without error, any member in a framed structure, it is probably best to assume the member to be cut, and to determine the external forces at each section which oppose the internal forces; the direction of the forces may then be determined with precision.

39. References for Graphic Statics:—For a more detailed study of graphic analysis reference may be made to Reuleaux, "The Constructor," pages 26-38; Church, "Mechanics of Engineering", pages 1-48; Low and Bevis, "Machine Design," pages 18-20; Supplee, "Mechanical Engineering", pages 234-258; Merriman and Jacoby, "Roofs and Bridges", Part II; Johnson, Bryan and Turneaure, "Theory and Practice of Modern Framed Structures", pages 11-32; Ketchum, "Steel Mill Buildings", pages 20-138.

40. Moment of a Force:—The moment of a force is *the product of the force and its lever arm*. If the force be measured in pounds and the lever arm in feet the moment will be in foot pounds. If the force be in pounds and the arm in inches the moment will be inch pounds. In most machine design calculations the latter would be used. When a system of forces acting on a body is in equilibrium, the sum of the moments acting in one direction about any given axis is equal to the sum of all the moments acting in the opposite direction about the same axis.

Illustration 1.—Given an imaginary beam having no weight and supporting a weight W as shown in Fig. 11. Find the reactions at A and B . Taking moments, first about B , and then about A , we have,

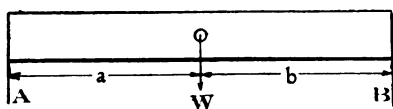


FIG. 11.

$$Wb = A(a+b)$$

$$A = \frac{Wb}{a+b} \quad (1)$$

$$Wa = B(a+b)$$

$$B = \frac{Wa}{a+b} \quad (2)$$

$$\text{Also note that } A + B = W. \quad (3)$$

Equation (3) combined with either (2) or (1) gives material for the complete solution of the beam.

Illustration 2.—Taking moments about A . Fig. 12, we have

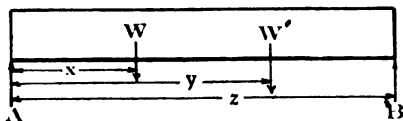


FIG. 12.

$$Bz = Wx + W'y$$

$$B = \frac{Wx + W'y}{z}$$

from which, if we remember that $W + W' = A + B$, we can get the solution of the beam.

NOTE:—If in the above the beam had weight, the bending moment due to this weight would in some cases be taken into account.

In most problems in Machine Design the forces acting upon any beam are so much greater than the weight of the beam that the latter is disregarded.

In every exact calculation, however, the weight of the beam should be considered. It will be desirable for the student to bear this in mind when the subject is discussed more fully later.

41. Load:—A load is a combination of external forces acting on a piece of a structure, or machine. Illustrations: A weight carried by a rope; a weight supported on a beam; power transmitted through shafting or belting, etc., etc.

Loads are classified according to action as *dead load*, *live load*, and *shock*. They are also classified as to distribution, as *concentrated* and *distributed*.

A *dead load* is one that is applied slowly and remains constant. A *live load* is one that is continually changing, but is not subjected to any sudden applications. *Shocks* are due to sudden applications and withdrawals usually alternating in opposite directions. The first and second can well be represented as the dead weight of the bridge, and the live load of the train passing over it. The third can be represented as the load on the piston rod, or the connecting rod of an engine. A *concentrated* load is one that is applied at one point and a *distributed* load is covered more or less evenly over the surface. Distributed loads are commonly *uniformly* distributed, as for example, the load due to the floor of a bridge, also the uniform weight of the floor supports.

42. Effect of a Load:—When a load is applied to a body it produces a change of form in that body. It also produces a corresponding stress on the fibres of the material. The change of form is sometimes known as *strain*, and is usually expressed as a certain amount per unit of length.

The internal force which is called into play to resist this deformation is called the *stress*, and is expressed in pounds per square inch. In machine design we are usually concerned with the stress and less often have to consider the strain.

Experiments conducted by a number of investigators serve to show that where we have a load repeatedly applied and removed, i. e., the equivalent of a live load, the breaking load is less than that required where the load is constant, that is, a dead load. The breaking load becomes less in the case of the repeatedly applied load by an amount proportional to the variation in stress produced. Thus, if a body is alternately subjected to tensile and compressive loads it will fail with a lighter load than when the load is simply applied and removed, producing but one kind of stress.

43. Kinds of Stresses:—There are essentially three kinds of stresses: *Tensional*, tending to elongate, *Compressional*, tending to

shorten, and *Shearing*, tending to cut the material across the grain or fibres.

In addition to these simple stresses, in most cases where the external forces are not directed along the axis of the piece, *flexure* or bending is developed and the resulting stress, which is a combination of compression and tension, is sometimes called *bending stress*. See Arts. 46 to 54. Also, when a body is subjected to the action of a force tending to rotate it, we call the stress developed a *torsional stress*, although the effect upon the body is that of a shearing stress. This resistance to twisting is fully considered in Arts. 66 to 73.

A combination of two or more of the foregoing general cases may occur, and in most instances, several of these stresses are produced at the same time.

The following equation gives the relation existing between the load and the stress, for the three elementary stresses, tension, compression and shear:—

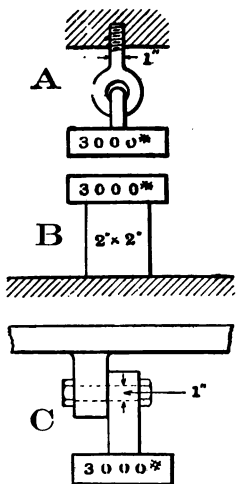
$$W = f A. \quad (4)$$

where W = load applied in pounds.

f = stress on the fibres in pounds per square inch.

A = cross sectional area of the material.

To illustrate the meaning of tension, compression and shear; we have first, a rod in tension supporting a weight Fig. 13. (A) where



$$W = f A.$$

$$3,000 = .7854 f$$

$$f = \frac{3,000}{.7854} = 3819.7 \text{ #1"}. \quad \text{Second, a block under pressure. (B) where}$$

$$W = f A.$$

$$3,000 = 4.f.$$

$$f = \frac{3,000}{4} = 750 \text{ #1"}. \quad \text{A bolt or pin under cross strain. (C) where}$$

$$W = f A.$$

$$3,000 = .7854 f$$

$$f = \frac{3,000}{.7854} = 3819.7 \text{ #1"}. \quad \text{FIG. 13.}$$

In applying the above general formula the following definitions should be kept well in mind:

The *ultimate stress* in any piece of material is the stress in pounds per square inch that the fibres are subjected to at the point of rupture. This factor differs in all materials.

The *working stress* is that stress in pounds per square inch to which a piece is to be subjected when it is a part of the machine. This is usually designated.

$$f = \frac{\text{ultimate stress}}{\text{factor of safety}}$$

44. Factor of Safety:—The factor of safety is the ratio of the ultimate stress to the working stress. This is assumed by the designer from his knowledge of the kind of work the machine will perform. In machine construction every piece sustains some sort of a load. If this load is a dead load the piece can be used under a greater fibre stress than if it be subjected to a shock, consequently the factor of safety will be smaller. Conversely, if the piece is under shock the factor of safety should be taken large so the working stress would be low.

It is sometimes confusing, for the person just taking up the subject of Machine Design, to understand why the factor of safety should vary through such wide limits in the same material when used under different conditions. It should be remembered, however, that all materials, when subjected to continued usage, become weaker and the ultimate strength, after much service, is less than it was at first. This is more noticeable in materials subjected to live loads and shocks than in those subjected to dead loads.

TABLE I.—Factors of Safety.

Loads and Stresses	Cast Iron	W't Iron	Steel
Shocks, and loads applied variably.....	20	15	12
Alternated stresses, of opposite sign.....	12	10	8
Equal, repeated stresses of one kind.....	10	8	6
Dead or non-variable loads.....	8	6	5

In designing any machine part for tension, compression or shear, it is first necessary to know the kind and amount of load which the part is to sustain. It is then necessary to know from the character of the material involved how many pounds per square inch it will stand before breaking, and assuming a factor of safety deter-

TABLE II.—Strength of Materials Most Common in Engineering Construction.

MATERIAL	Per Cent of Carbon	Tensile Strength		Compressive Strength		Shearing Strength	Flexure	Modulus of Elasticity		Shock Resistance	Methods of Shaping for Use
		Elastic Limit	Ultimate	Elastic Limit	Ultimate	Ultimate	Stress in Outer Fibre	Tension	Shear		
Cast Iron....	1.5 to .4		$\left\{ \begin{array}{l} 10000 \\ \text{to} \\ 35000 \\ \text{aver.} \\ 20000 \end{array} \right\}$	$\left\{ \begin{array}{l} 50000 \\ \text{to} \\ 140000 \\ \text{aver.} \\ 90000 \end{array} \right\}$	$\left\{ \begin{array}{l} 12000 \\ \text{to} \\ 25000 \end{array} \right\}$		$\left\{ \begin{array}{l} \text{Ultimate} \\ 30000 \\ \text{to} \\ 54000 \\ \text{aver.} \\ 42000 \end{array} \right\}$	12000000	5000000	Low	Casting.
Steel Castings....	.1 to .75	40000	60000 to 90000	40000	125000	60000	700000	20000000 to 30000000 aver. 25000000		Good	Casting.
Malleable Cast Iron.....		16000	35000			42000	Ultimate 70000			High	Casting.
Crucible or Tool Steel.....	.4 to .15	65000	120000	65000				35000000	13000000	Medium	Rolling and forging.
Common Wrought Iron.....	.1 to .5	22000	40000	22000	48000	35000		30000000	10000000		
High Grade Wrought Iron..		28000	56000	28000	50000	45000		28000000	10000000	High	
Bessemer and Open Hearth Steel, also called Machinery Steel, Mild Steel and Ingot Steel	0.15 to 0.96	42000 to 68000	63000 to 115000	39000 to 71000	60000 to 100000	48000 to 83000	60000 to 120000		30000000	High to Low	Rolling, forging and wire drawing.
Nickel Steel, Oil Tempered.....		65000	100000	65000	100000	50000		30000000	12000000	High	Casting, rolling, forging and wire drawing.
Nickel Steel, Annealed.....		50000	90000	50000	90000	50000		80000000	12000000	High	Crankshaft forging
Vanadium Steel...	.26	50000	75000					30000000	12000000	Very High	
Brass.....		13000	25000 to 50000 aver. 35000		12000		16000	9000000			Casting, rolling, forging and wire drawing.
Bronze.....		25000	30000 to 60000 cast		70000	40000	30000	12000000			Casting, rolling, forging wire drawing.
Copper.....						20000 to 30000		15000000			Casting.
Aluminum.....		14000	28000		13000			11000000			

mine the working stress, or fibre stress f in the formula $W = f A$. This gives the area of the piece necessary to fill the conditions.

45. Table II gives accepted values for the fibre stresses of the materials commonly used in machine design. It will be noticed that the figures given in some cases vary through wide limits; this is due to the difference in texture between different pieces of the same class of material. The designer should take especial notice of this in making his selection.

46. **Columns:**—Any piece of material in compression, having a length of 15 or more times its least cross sectional dimension, is called a column or pillar. Ordinary rules for compression may not apply in such a case.

The stiffness of a column varies almost inversely as the square of its length, i. e. if a column is designed to sustain a load of 10,000 pounds, and its length be increased twice, it will carry approximately $10,000 \div 4 = 2500$ pounds safely, providing the cross sectional areas remain the same.

The *condition of the ends* of a column affect its stiffness, i. e., its capacity to resist buckling. In Fig. 14, A is *square ended*; B is *pin and square ended*; and C is *round ended*. A may be either flat, as shown, or fixed, as in D, E, and F. C may take the forms of G, H or I; although H is regarded as square ended, if buckling is considered as taking place in a plane perpendicular to the page. Flat ends should be well fitted, otherwise they should be classified as round.

When the length of compression member is more than 15 or 20 times its least cross sectional dimensions, it is well to investigate for buckling. The following formulas by *Rankine* give the breaking loads P , P' and P'' and apply fairly well to columns of any length.

$$A. \text{ Square ends. } P = \frac{C A}{1 + a \frac{l^2}{k^2}} \quad (5)$$

$$B. \text{ Pin and square. } P' = \frac{C A}{1 + \frac{1}{4} a \frac{l^2}{k^2}} \quad (6)$$

$$C. \text{ Round. } P'' = \frac{C A}{1 + 4 a \frac{l^2}{k^2}} \quad (7)$$

P , P' and P'' are the breaking loads (afterwards to be reduced by the factors of safety); A , the least area of cross section; C , the modulus of crushing (ultimate unit crushing strength); a , a constant given in the following table; l , the length of the column in inches; and k the radius of gyration of the section, in

inches. The radius of gyration of any section may be obtained by taking the square root of the quotient obtained by dividing the moment of inertia by the area of the section. Thus if r be

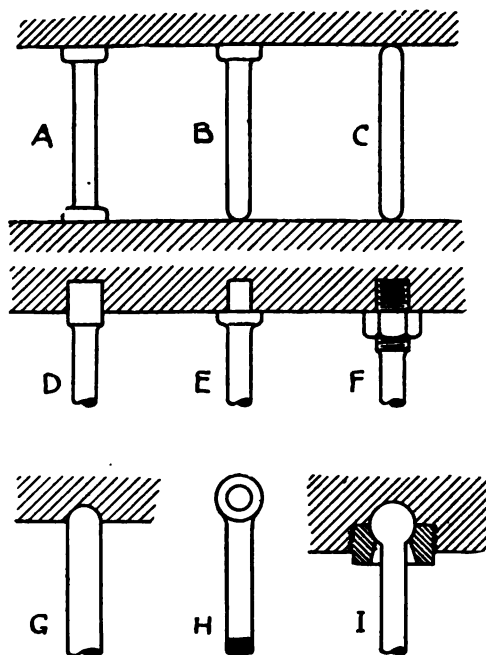


Fig. 14.

the radius of gyration, I the moment of inertia and A the area of the section we have

$$r = \sqrt{\frac{I}{A}}$$

TABLE III.

Solid Sections	Cast Iron	Wrought Iron	Soft Steel	Medium Steel	Hard Steel
C , lbs. per sq. in. =	70000	36000	45000	50000	70000
a =	$\frac{1}{6400}$	$\frac{1}{36000}$	$\frac{1}{36000}$	$\frac{1}{36000}$	$\frac{1}{25000}$

For columns having a length of, say, 150 times the least radius of gyration, the following, known as *Eulers* formulas will give satisfactory results.

$$A. P = 4 E I \pi^2 \div l^2$$

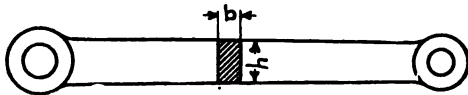
$$B. P' = \frac{3}{4} E I' \pi^2 \div l^2$$

$$C. P'' = E I'' \pi^2 \div l^2$$

Where P , P' , P'' and l are as stated above; E , the modulus of elasticity; and I , I' and I'' , the moments of inertia of the section. From these formulas, with the same cross-section in each column, if the strength of $C = 1$, then $B = \frac{3}{4} C$. Also, for the same load, if the length of $C = 1$, then the length of $B = 1\frac{1}{2}C$ and $A = 2C$, for columns of the same rigidity.

Connecting rods usually have a rectangular cross section Fig. 15. Such a rod is a combination of A . and C . It can be shown, as follows, that the height h should be twice the breadth b to be equally strong to resist buckling in either plane. To fulfill

this condition $P = P''$ or $4 E I \frac{\pi^2}{l^2} = E I'' \frac{\pi^2}{l^2}$, $4 I = I''$, then $\frac{4 b^3 h}{12} = \frac{b h^3}{12}$ and $h = 2b$.



In applying the column formulas to machine parts, it is well to first design the parts to resist compression, and then, see if they are rigid enough to resist buckling.

Application.—A sliding ram is connected to the driving shaft of a machine by a rectangular sectioned, mild steel connecting rod 36 inches long. This rod carries 100,000 pounds compressive stress. How large must be the section of the rod to resist compression and buckling?

Assuming this to be a slow moving machine, the safe compressive stress may be taken at 10,000 pounds per square inch. The area of the section then, is 10 square inches. With $h = 2b$, the section of the rod will be 2.25×4.5 inches. In this case the relation existing between the length and least cross sectional dimension is such that Rankine's Formula should be used; and, if the rod is equally strong in both planes, either formula A , or C (Rankine) may be applied. Taking the latter we have,

$$P'' = \frac{CA}{1 + 4a \frac{l^2}{k^2}} = \frac{50000 \times 10}{1 + \frac{4}{36000} \frac{(36)^2}{1.69}} = 717500 \text{ pounds.}$$

This shows that the rod has a factor of safety of $717500 \div 100000 = 7 +$ for buckling, and 5 for compression.

47. References for Columns.—Merriman, "Mechanics of Materials," pages 111-134; Church, "Mechanics of Engineering," pages 352-385.

48. Bending:—Another kind of strain and stress that is very common in machine construction is *bending*. This deals with the moments of forces rather than with simple pressures, and the formula is:

$$M = f Z. \quad (8)$$

where M = bending moment (usually given in inch pounds.)

f = fibre stress in pounds per square inch.

$Z = \begin{cases} \text{Modulus of the section.} \\ \text{Résistance of the section,} \\ \text{or section factor.} \end{cases}$

This formula will be continually used and should be well understood. To make the explanation clear, assume a cantilever beam as in Fig. 16, projecting from a wall and acted upon by a weight W a distance l inches from the point of support. It is evident that the

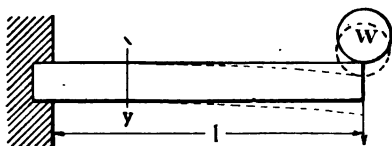


FIG 16.

beam will be under a strain as shown by the dotted lines, with the upper fibres in tension, the lower fibres in compression and the neutral axis free from stress of any kind. It will also be understood that the fibre stress at any section

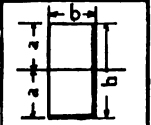
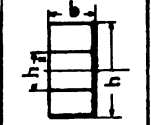
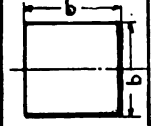
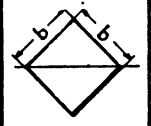
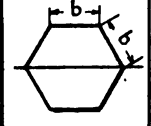
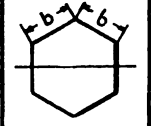
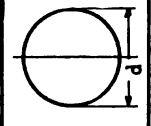
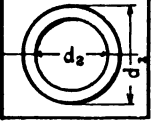
$x y$ will be in a direct ratio to the distance from the neutral axis, consequently, it will follow that certain portions of the section will be under more stress than other portions. From this point of view the resistance of the section will not be in proportion to the area of the section, but to some other factor (Z) called the *modulus of the section*, or *section factor*. The modulus is found by dividing the moment of inertia of the section by the distance from the center of gravity to the outermost fibre. (See Church's Mechanics, Par. 90) Z will be different for different shaped sections. In the reference as given the moment of inertia of a rectangular section about its gravity axis, is $b h^3 \div 12$. The distance from the gravity axis to the outermost fibre is $h \div 2$ where h is the height of the beam section, consequently, $2b h^3 \div 12 h = b h^2 \div 6$ which is the value of Z in the formula, if the beam has a rectangular section.

Suppose the beam is circular in section, the modulus from Par. 91, Church, becomes

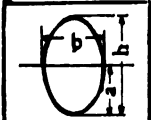
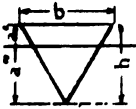
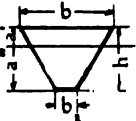
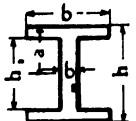
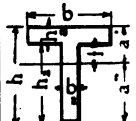
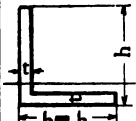
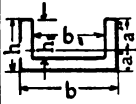
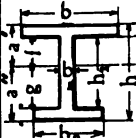
$$\frac{\pi r^4}{4r} = \frac{\pi}{4} \left(\frac{d}{2} \right)^3 = \frac{\pi d^3}{32}$$

49. References for Bending:—Lanza, "Applied Mechanics," pages 267-275; DuBois, "Stresses in Framed Structures," pages 282-321. Church, pages 244-385.

SECTION TABLE IV. Modulus for Bending.

SECTION	MODULUS	AREA
	$\frac{b h^3}{6}$	$b h$
	$\frac{b (h^3 - h_1^3)}{6 h}$	$b (h - h_1)$
	$\frac{b^3}{6}$	b^2
	$\frac{\sqrt{2}}{12} b^3 = 0.118 b^3$	b^2
	$\frac{5}{8} b^3$	$\frac{3\sqrt{3}}{2} b^2 = 2.598 b^2$
	$\frac{5\sqrt{3}}{16} b^3$	$\frac{3\sqrt{3}}{2} b^2$
	$\frac{\pi d^3}{32}$	$\frac{\pi d^2}{4}$
	$\frac{\pi}{32} \left(\frac{d_1^4 - d_2^4}{d_1} \right)$	$\frac{\pi}{4} (d_1^2 - d_2^2)$

SECTION TABLE IV. Modulus for Bending—Continued

SECTION	MODULUS	AREA
	$\frac{\pi}{32} b h^3$	$\frac{\pi b h}{4}$
	$\frac{b h^2}{12} = Z'$ $\frac{b h^2}{24} = Z''$	$\frac{b h}{2}$
	$\frac{b^2 + 4bb_1 + b_1^2}{12(b + 2b_1)} h^2 = Z'$ $\frac{b^2 + 4bb_1 + b_1^2}{12(2b + b_1)} h^2 = Z''$	$\frac{b + b_1}{2} h$
	$\frac{bh^3 - (b - b_1)h_1^3}{6h}$	$bh - (b - b_1)h_1$
	$\frac{b(a'^3 - f^3) + b_1(f^3 + a''^3)}{3a'} = Z'$ $\frac{b(a'^3 - f^3) + b_1(f^3 + a''^3)}{3a''} = Z''$	$b_1 h_1 + bh_2$
	even angle $\frac{A h}{7.2}$	$ht + (b - t)t$
	$Z_{a'} = \frac{2A[ba'^3 + b_2a''^3 - (b - b_1)f^3 - (b_2 - b_1)g^3]}{3[b_1h^2 + (a' - f)^2(b - b_1) + (ta'' - g)(b_2 - b_1)(2h - a' + g)]}$	
	$Z_{a'} = \frac{2A[(b - b_1)h^3 + b_1(h - h_1)^3] - 6A^2a'^2}{3[h^2(b - b_1) + b_1(h - h_1)^2]}$	

50. Section Modulus:—No attempt will be made in the early part of the work to figure out the values of the modulus Z for different sections since this is the province of the mechanics, but values of Z are given in Table IV for all common forms of sections.

51. Comparative Strength of Sections:—To return to the formula $M = fZ$ again; the value of f is the same as that used in (4) and M is the bending moment expressed always in terms of inch pounds, as, $M = Wl$, where W = weight in pounds, and l = length of the lever arm in inches.

Application.—Having given a beam as in Fig. 17, where $W = 3600$ pounds and $l = 36$ inches; find the maximum size of beam section that would be used in each of the following conditions: first, square section set flat; second, square section set diagonally; third, rectangular section; fourth, circular section; fifth, equilateral triangular section. Assume mild steel for each and $f = 8,000$ then

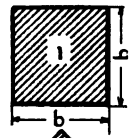
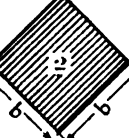
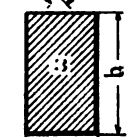
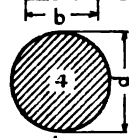
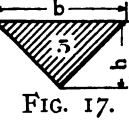
(1)		$M = fZ \dots \dots \dots \text{Area of section} = 20.16 \text{ in}^2$ $129600 = \frac{8000 b^3}{6} \dots \dots \text{Total shear } S = 3600 \text{ \#}$ $b = 4.6'' \dots \dots \dots \text{Unit shear } fs = 178 \text{ \#in}''$
(2)		$M = fZ \dots \dots \dots A = 27.04 \text{ in}^2$ $129600 = 8000 (.118 b^3) \dots \dots \dots S = 3600 \text{ \#}$ $b = 5.2'' \dots \dots \dots fs = 133 \text{ \#in}''$
(3)		$M = fZ \dots \dots \dots A = 17.1 \text{ in}^2$ $129600 = \frac{8000 b h^2}{6} \dots \dots \dots S = 3600 \text{ \#}$ assume $b = 3''$ then $h = 5.7'' \dots \dots fs = 210 \text{ \#in}''$ If $b = 2''$ then $h = 7''$ and the area becomes 14 in^2 instead of 17.1 in^2 .
(4)		$M = fZ \dots \dots \dots A = 23.75 \text{ in}^2$ $129600 = \frac{8000 \pi d^3}{32} \dots \dots \dots S = 3600 \text{ \#}$ $d = 5.5'' \dots \dots \dots fs = 151 \text{ \#in}''$
(5)		In the triangular section we have the modulus for the base $\frac{b h^2}{12}$ and that for the point $\frac{b h^2}{24}$. Taking $Z = (1 \div 24) b h^2$, since this is the weakest point we have $M = fZ \dots \dots \dots A = 28.4 \text{ in}^2$ $129600 = \frac{8000 b h^2}{24} \dots \dots \dots S = 3600 \text{ \#}$ $b h^2 = 388.8 \dots \dots \dots fs = 137 \text{ \#in}''$. but $h = b \cos 30^\circ = .866 b$. $b = 8.1''$

FIG. 17.

The first four beams were symmetrical in section, that is, the gravity axis passed through the center of the section, as shown in 10 of the 16 sections of the modulus table, but when the section area is not uniform as in a tee bar or triangle where the gravity axis is above or below the center of the height there is a modulus for each part. The smaller of the two is usually taken as the working modulus. See Par. 90, a, Church's Mechanics.

The foregoing will serve to show the use of the modulus in figuring beams. It will be noticed that the rectangular beam requires the least weight of any of the beams considered. It will also be seen that the strength of the rectangular beam varies directly as the breadth, and as the square of the height. If the breadth of the beam be doubled the strength is doubled, the weight is also doubled, but if the height is doubled the weight is doubled and the strength is quadrupled. It is an advantage to have beams as light as possible for any given strength, and by the above method a selection can easily be made. If this problem were tried in comparison with the rolled steel forms such as the I beam the latter would give even better results.

52. References for Section Modulus:—Lanza, "Applied Mechanics," pages 275-283; "Cambria Steel," pages 144-153; Supplee, "Mechanical Engineering," pages 353-359; DuBois, "Stresses in Framed Structures," pages 270-282; Reuleaux, "The Constructor," pages 3-8.

53. Maximum Bending Moment:—It will be necessary at times for the designer to locate the point in a beam where the maximum bending moment takes place. In a simple beam having two supports and loaded uniformly, or at the middle, it is located at the center of the beam while in a cantilever it is at the point of support. *The point of maximum moment in any case will be where the shear changes from positive to negative and vice versa.*

Application:—Having a beam loaded as shown in Fig. 18, we find the reactions at *A* and *B* to be 878.4 and 721.6 pounds respectively. Now the shear diagram would be as follows: from *A* to *C* 878.4 pounds; from *C* to *D* $878.4 - 500 = 378.4$ pounds; from *D* to *E* $378.4 - 1000 = -621.6$ pounds; from *E* to *B* $-621.6 - 100 = -721.6$ pounds. Hence according to our former statement the greatest bending moment is at *D*. The value of this bending moment will be $878.4 \times 95 - 500 \times 45 = 60948$ inch pounds.

The bending moment may be scaled directly from the equilibrium polygon, or moment diagram, Fig. 18, whose ordinates give the bending moments in the corresponding sections of the beam. It will be noticed that the *maximum* ordinate occurs at the meeting point between the sides of the equilibrium polygon that are parallel to *o2* and *o3*, the rays, in the force polygon on opposite sides of and adjacent to *oh*, the line parallel to the closing side, *mn*, of the equil-

ibrium polygon. The maximum bending moment of the cantilever beams and the simple beams shown in Table V, bring in only the simplest considerations. Other and more complicated loadings can be looked up in Church's *Mechanics of A* Engineering Pars. 239-290; Cotterill's *Applied Mechanics*, Pars. 28-45, 153-172; Perry's *Applied Mechanics*, Pars. 349-367.

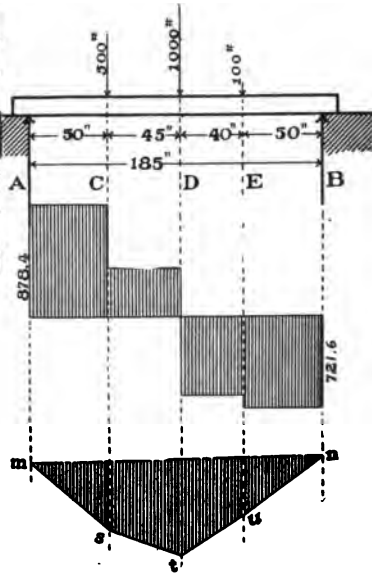
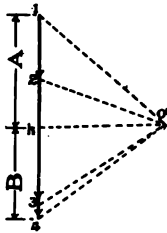


FIG. 18.

54. Shear:—The shearing load at any section in a beam is equal to the resultant of all the parallel forces acting on the beam on one side of that section. To illustrate, Fig. 19 gives the shear at a , b , c , d , and e $f = W$ for a cantilever, while for a simple beam the shear at

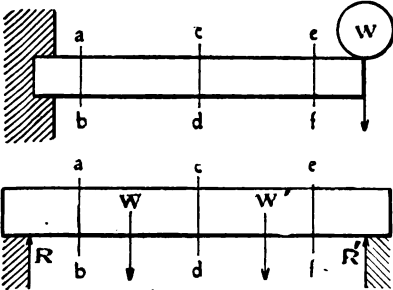


FIG. 19.

signed sufficiently strong to resist bending it generally fulfills all the requirements for shearing. Notice the values of f for flexure and shear as obtained in the application Fig. 17.

In calculating the shear of an I beam it is customary to assume that the entire shear is taken up by the web, as $b' w$, Fig. 20. See Church, Par. 256. Knowing this area and the total shear, the value f_s can be obtained.

$a b$ is $R = W + W' - R'$
 $c d$ is $R - W = W' - R'$
 $e f$ is $R - (W + W') = R'$

For the construction of the shear diagram for a simple beam reference should be made to Art. 53, Fig. 18.

A beam may fail either by an excess of bending or by shearing. The former is usually the case however. If a beam is de-

55. Beams Classified:—Beams are classified as simple, cantilever, restrained and continuous.

A *simple beam* is one which rests upon two end supports, as IV to X inclusive, Table V.

A *cantilever beam* rests upon a support at the middle, or has one end fixed and the other end free, as I to III inclusive, Table V.

A *restrained beam* or *fixed beam*, has both ends fixed, as XI and XII, Table V.

A *continuous beam* is one which rests upon more than two supports.

Wooden beams are usually rectangular in section while steel and wrought iron beams are generally as shown in Fig. 20.

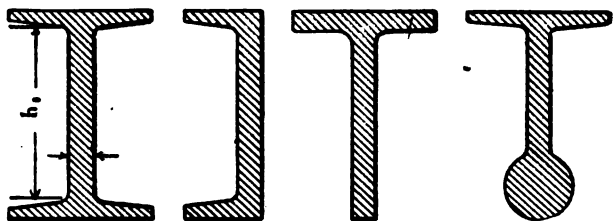


FIG. 20.

The various rolled beams are distinguished by *inches in depth of section* and *pounds per foot* of length as, 15" 80# beam.

56. Relative Strength of Beams:—Table V gives the relative strength of beams carrying uniformly distributed or concentrated loads. This table gives the most common forms of the applications of loads to beams. For a continuation of the table as applied to unusual sections refer to

The Constructor	Pages 3- 5
Unwin, (Vol. I.)	Pages 54-72
Low & Bevis	Pages 34-35

57. Deflection of Beams:—In some lines of machine design it is not necessary to determine the amount of deflection in the various members of a machine, in others, however, it would be absolutely necessary, as for example, in all forms of testing machines. The formulas given in Tables V for deflection may be applied in such cases.

In general, the beam member would be calculated in section for strength, first, and then the deflection formulas applied to see if the section will give sufficient rigidity. In some cases, a beam may be desired of such a size as will allow of only a certain deflection; the deflection formula would then be applied first, to obtain I , the moment of inertia of the section, from which the shape of the section would be obtained.

TABLE V PART I




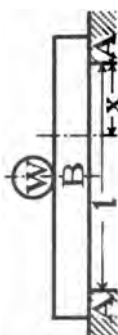
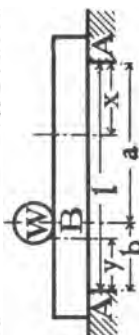
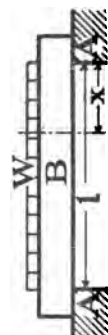
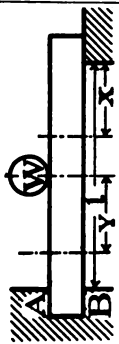
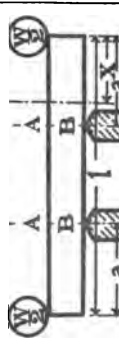
Example. Lengths in inches; Loads in pounds	Type of Beam. Mode of Loading.	Bending Moment in inch-pounds	Max. B'n'd'g Mom., M.	Max. Deflection, d, in inches	Max. Shear, at A, in lbs.	Weakest Section at
	I Cantilever Beam Concentrated Load at free end	Wx	Wl	$\frac{Wl^3}{3EI}$	W	B
	II Cantilever Beam Uniform Load	$\frac{Wx^2}{2l}$	$\frac{Wl}{2}$	$\frac{Wl^3}{8EI}$	W	B
	III Cantilever Beam Distributed Load in- creasing uniformly toward fixed end	$\frac{Wx^3}{3l^2}$	$\frac{Wl}{3}$	$\frac{Wl^3}{15EI}$	W	B
	IV Simple Beam Concentrated Load at center	$\frac{Wx}{2}$	$\frac{Wl}{4}$	$\frac{Wl^3}{48EI}$	$\frac{W}{2}$	B
	V Simple Beam Concentrated Load at any point	For right side $\frac{Wbx}{l}$ For left side $\frac{Way}{l}$	$\frac{Wab}{l}$	$\frac{\sqrt{3W^2a^3b^2(2l-a)^3}}{27EI}$	$\frac{Wb}{l}$ $\frac{Wa}{l}$	B
	VI Simple Beam Uniform Load	$\frac{Wx}{2} \left(1 - \frac{x}{l}\right)$	$\frac{Wl}{8}$	$\frac{5Wl^3}{384EI}$	$\frac{W}{2}$	B

TABLE V PART II

Example. Lengths in inches; Loads in pounds	Type of Beam. Mode of Loading.	Bending Moment in inch-pounds	Max. B'n'd'g Mom., M.	Max. Deflection, d, in inches	Max. Shear, at A, in lbs.	Weakest Section at
	VII Simple Beam Distributed Load in- creasing uniformly toward one end	$\frac{Wx}{3} \left(1 - \frac{x^2}{l^2} \right)$	$\frac{104 Wl}{810}$	$\frac{47 Wl^3}{3600 EI}$	$\frac{2W}{3}$	$x=0.52 l$
	VIII Simple Beam Distributed Load in- creasing uniformly toward center	$Wx \left(\frac{1}{2} - \frac{2x^2}{3l^2} \right)$	$\frac{Wl}{6}$	$\frac{Wl^3}{60 EI}$	$\frac{W}{2}$	B
	IX Simple Beam Distributed Load in- creasing uniformly toward ends	$Wx \left(\frac{1}{2} - \frac{x}{l} + \frac{2x^2}{3l^2} \right)$	$\frac{Wl}{12}$	$\frac{3 Wl^3}{320 EI}$	$\frac{W}{2}$	B
	X Simple Beam Two Symmetrical Loads	$\frac{Wx}{2}$	$\frac{Wa}{2}$	$\frac{Wa}{48 EI} (3l^2 - 4a^2)$	$\frac{W}{2}$	B, B,
	XI Fixed Beam Concentrated Load at center	$\frac{Wl}{2} \left(\frac{x}{l} - \frac{1}{4} \right)$	$\frac{Wl}{8}$	$\frac{Wl^3}{192 EI}$	$\frac{W}{2}$	B, B, B,
	XII Fixed Beam Uniform Load	$Wl \left(\frac{x}{l} - \frac{x^2}{l^2} - \frac{1}{6} \right)$	$\frac{Wl}{12}$	$\frac{Wl^3}{384 EI}$	$\frac{W}{2}$	B, B,

TABLE V PART III

Example, Lengths, in inches; Loads in pounds	Type of Beam, Mode of Loading	Bending Moment in inch-pounds	Max. B'n'd'g Mom., M.	Max. Deflection, d, in inches	Max. Shear, at A, in lbs.	Weakest Section at
	XIII Combination Simple and Fixed Beam Concentrated Load at center	For right side $\frac{5Wx}{16}$ For left side $\frac{5}{16}Wl$ $Wl\left(\frac{3}{32} - \frac{x}{16l}\right)$	$\frac{3Wl}{16}$	$\frac{3Wl}{32EI}$	$\frac{11W}{16}$	B
	XIV Combination Simple and Fixed Beam Uniform Load	$\frac{Wx}{2}\left(\frac{3}{4} - \frac{x}{l}\right)$	$\frac{Wl}{8}$	$\frac{5Wl^3}{926EI}$ For overhang $\frac{Wa}{12EI}\left(3a - 4a^2\right)$ Between supports $\frac{Wa}{16EI}\left(l - 2a\right)^2$	$\frac{5W}{8}$	B
	XV Two Symmetrical Supports Concentrated Load at each end	$\frac{Wx}{2}$	$\frac{Wa}{2}$		$\frac{W}{2}$	B, B,
	XVI Two Symmetrical Supports Uniform Load	$\frac{Wx}{2}\left(\frac{x}{l} - 1 + \frac{a}{x}\right)$ At center of span $\frac{W}{2}\left(a - \frac{l}{4}\right)$	At either support $M = \frac{Wa^2}{2}$ The supporting power is Max. when $a = 0.207l$, in which case Max. $M = \frac{3Wl}{140}$		$\frac{W}{2}$	B, B,

58. Selection of Beams:—It is very convenient in specifying beams of the standard rolled sections to refer to such books as Cambria Steel, Jones & Laughlin, and Carnegie, for details of sections, section modulus, safe loads and spacing.

Illustration.—Wanted an I beam to support a ten ton moving load Fig. 21, with an allowable fibre stress in the beam of 16000 pounds per square inch. Consulting Cambria 1909 we find that all

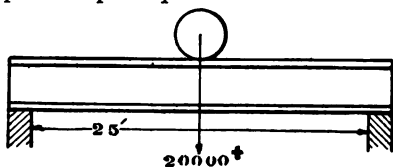


FIG. 21.

figures refer to uniformly loaded beams, however, from the preceding table it shows that a simple or cantilever beam uniformly loaded will stand twice as much load as a beam of the same span having a concentrated load. This fact permits the application of a uniformly loaded beam from Cambria carrying a load of 40,000 lbs. Page 90 gives a special B-109, 15", 80 lb. beam; also on page 92 a standard B-65, 18", 65 lb. beam which will fulfill the same conditions. Refer to page 168 for the properties of the standard beam and then to Page 6 for the shape and size of the section. Having obtained the area of the web the value f_s may be calculated. In the above case the greatest shear comes on the beam when the load is at the support then $S = 20,000$ lbs. From the section the area of the web of B—65 is 7.434 square inches which gives $f_s = 2690$ pounds per square inch.

It should be noted that the fibre stress in this case was used much higher than would be ordinarily used in the designing of machine parts. Suppose some other fibre stress as 10,000, is desired as a working stress then the total load would be computed and increased by the ratio of the two fibre stresses. The beam would then be selected for this load. To illustrate: 20,000 pounds concentrated at the center or 40,000 pounds uniformly distributed will produce a fibre stress of 16,000 lbs. sq. in. in some standard section, then if we select a beam that will support $40,000 \times 16 \div 10 = 64,000$ pounds at a fibre stress of 16,000 lbs. sq. in. it will support 40,000 lbs. at a fibre stress of 10,000 lbs. sq. in. Such a beam as found in Cambria, page 93 gives, No. B. 121, 20 in. Special 85 lbs.

This can be shown in another way:—

$\frac{Wl}{4} = f Z; \frac{Wl}{4f} = Z = 150$, which is the modulus for the above beam as shown on Page 168, Cambria.

It is sometimes desirable to know how far a beam will deflect from its normal state, when loaded, to see if its lack of stiffness will endanger the materials carried by the beam.

In Machine Design only slight deflections are permissible, while in bridge or structural designs greater allowances are made. Cambria gives for plastered ceilings $\frac{1}{8}$ of the span as a maximum allowable deflection for beams supported at the ends and carrying lath and plaster ceilings.

Having given an *allowable* deflection, the deflection formula is applied and the result compared with the allowable deflection.

Application 1. A B-9, 4", 8.5 lb. I-beam from Cambria, page 166, is projecting 24 inches from the side of a machine frame and loaded at the end with 1600 pounds. A deflection of 0.05 inches is allowable; will the beam be satisfactory? From Table V we have

$$d = \frac{1600 \times (24)^3}{3 \times 30,000,000 \times 6.4} = 0.038 \text{ inches.}$$

which shows that the beam will have sufficient stiffness.

Application 2. Given the same beam to carry as large a load as possible so that the deflection does not exceed 0.05 inches. Applying the same formula, $W = 2083$ pounds as the maximum allowable load.

Application 3. Assume the same conditions as in Fig. 21, with $f = 10,000$ pounds per square inch, and find the deflection of the Special I-Beam, 20 inch No. B 121, 85 lbs. From Table V,

$$d = \frac{20000 \times (25 \times 12)^3}{48 \times 30,000,000 \times 1508.5} = 0.248 \text{ inches.}$$

As a check upon such work look up Cambria Page 76.

59. References for Beams:—Lanza, "Applied Mechanics", pages 743-778; "Cambria Steel," pages 144-150; Church, "Mechanics of Engineering," pages 320-332, 484-514; Merriman, "Mechanics of Materials", pages 36-110; DuBois, "Stresses in Framed Structures", pages 171-189, 340-361.

60. Problems:

(1). It took 67500 pounds tensional pull to break a wrought iron rod $1\frac{1}{4}$ inches in diameter. What was the ultimate strength in pounds per square inch?

(2). An air hoist works with 80 pounds air pressure per square inch. The piston is 8 inches in diameter. Find the diameter of the piston rod to lift the maximum load, allowing 10 per cent. for friction; the working stress to be 4000 pounds per square inch.

(3). A 10 inch x 12 inch cylinder of a steam engine uses steam at 100 pounds gage pressure. The head is held in place by 6 bolts. If the working strength is to be 3500 pounds per square inch, find the diameter of the bolts. *Note:*—The effective area of a bolt is the area at the root of threads.

(4). A rod 1 inch in diameter and 5 inches long, is under a compression load of 6000 pounds. What is the unit stress?

(5). The piston of a steam engine is 20 inches in diameter; the steam pressure by gage is 49.5 pounds per square inch; the piston rod is of steel, $1\frac{3}{4}$ inches in diameter, and 30 inches long. Considering the rod to act in compression, find the working stress in the piston rod in pounds per square inch; also find the working stress in the rod when considered as a square ended column.

(6). The pressure on a punch, punching a $\frac{3}{4}$ inch hole in a wrought iron plate $\frac{5}{8}$ inches thick, is 78000 pounds. Determine the ultimate shearing stress of the wrought iron.

(7). A wrought iron bolt $1\frac{1}{2}$ inches in diameter is suspended through a hole of the same size in a mild steel plate $1\frac{1}{4}$ inches thick, and is prevented from dropping through the hole by the head of the bolt which is 3 inches square and $\frac{3}{4}$ inches thick. If the bolt supports a weight of 3000 pounds at the lower end, find the tension in the bolt in pounds per square inch; find the force tending to shear the head in pounds per square inch; and find the compressional force acting upon the lower face of the head of the bolt in pounds per square inch.

(8). Required the safe working load for a standard 4 inch wrought iron pipe used as a long column, having a length of 12 feet, and an external diameter of $4\frac{1}{2}$ inches, allowing a factor of safety of 6 in each of the cases a, b and c.

(a). The ends well fitted with flat bases.

(b). Flat base at lower end with ball-and-socket bearing at upper end.

(c). Pin connected at both ends.

(d). Examine the pipe for the largest load and see if it is safe from straight compression.

(e). What would be the safe working load for a solid cast iron column, diameter $4\frac{1}{2}$ inches, under the above conditions, (a), (b), and (c)?

(9). A cantilever beam, rectangular in section, is 8 feet long and 9 inches deep, and has a safe working load of 5000 pounds placed at its end; having given that the breaking load at the center of a simple beam of the same material, 9 inches long and 1 inch diameter is 600 pounds; it is required to find the breadth of the section. Find also the breadth when an additional load of 2 tons is distributed uniformly over the length. Factor of safety in each case 6.

(10). A cantilever beam is 12 feet long, 3 inches thick, and 5 inches deep, and bears a uniformly distributed load of 2 tons. Find the depth of a beam supported at each end, length 16 feet, breadth 6 inches, having a center load of 12000 pounds and the same fibre stress.

(11). Determine the dimensions of the strongest and stiffest beam that can be cut from a log of timber 1 foot 4 inches in diameter.

(12). A beam 15 feet long supports 5500 pounds at its center. If this load is replaced by two other loads, each placed 4 feet from an end, find the total weight of the two loads so that the maximum stress on the beam may be the same as under the original loading.

(13). A beam supported at the ends is b inches broad and d inches deep; it supports W pounds at the center. If W be moved to half-way between center and end, how much may d be reduced?

(14). Select a Standard or Special I-Beam for a five ton traveling crane, 30 feet span. Assume the working strength to be 9000 pounds per square inch. Two beams are used in the construction, each taking half the load. What will the maximum shear in the web be? Determine the deflection of the beam.

CHAPTER III.

Notes and Formulas Continued.

61. Work:—Work is the result of force in motion and can be defined as *the overcoming of resistance along a line of motion*. Illustrations: A weight of 200 pounds is lifted 10 feet; the work done is 2000 foot-pounds. A locomotive has an average draw-bar pull in one mile of 2,000 pounds; the work done is 10,560,000 foot-pounds.

Work is independent of time.

62. Power:—Power is the *rate of doing work*. Suppose the 200 pound weight was lifted 10 feet in one second of time, the power exerted would be 2000 foot-pounds per second. The term power is usually stated as *horse-power*, the equivalent of which is 33,000 foot-pounds per minute; the above illustration would then be $2000 \times 60 \div 33,000 = 3.6$ H. P.

The relation between power and work, expressed as a formula is

$$\text{H. P.} = \frac{P \cdot V}{33,000} \quad (1)$$

Where P = force in pounds and V = velocity in feet per minute.

Formula (1) is the standard horse-power formula and can be used as a basis for all power work.

63. Energy:—The kinetic energy, K. E. of a body is the *ability of that body to do work*. This quality expressed in terms of a formula is

$$\text{K. E.} = \frac{W v^2}{2g} \quad (2)$$

where W is the weight of the body in pounds, v is the velocity in feet per second and $g = 32.2$.

64. Torsion:—The last of the five applications of forces commonly met with in practice is torsion and since it relates mostly to the twisting of shafts the discussion of it was reserved until after mentioning the subject of Power.

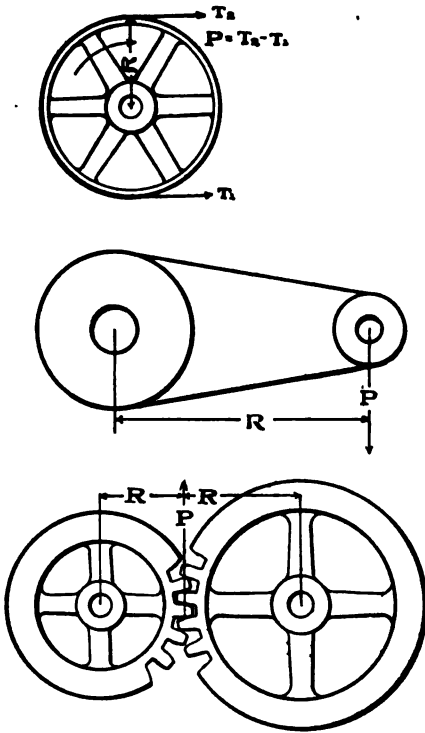


FIG. 22.

It will be noticed that the moment $P. R.$ depends only upon the H. P. and the R. P. M. The same relation shown above will be true for the crank and the gears if $P.$ be constant.

65. Twisting Moment:—

Fig. 22 shows three applications of the twisting moment in machines; the pulley, the crank and the gears. In the case of the pulley let the belt while transmitting power have the two tensions T_2 and T_1 , then the effective turning force of the belt is $P = T_2 - T_1$. This acts through a lever arm R , usually given in inches, giving a twisting moment $P. R.$ The value $P. R.$ is obtained directly from formula (1) as follows:

$$\text{H. P.} = \frac{P. V}{33000} = \frac{2 \pi R. N. P.}{33000 \times 12}$$

$$P. R. = \frac{33000 \times 12 \times \text{H. P.}}{2 \pi N.} \quad (3)$$

$N.$ = Revolutions per minute.

Shafting and Shafting Supports.

66. Resistance of a Shaft to Torsion:—Any shaft subjected to a twisting moment has its fibres subjected to a shear. Imagine two planes AB and $A'B'$ Fig. 23, in contact and perpendicular to the center line of the shaft. If the shaft be held from turning and a force P be applied to the crank the tendency will be to turn one plane section on the other and the two lines r, s , which coincided before the application of the force, now take some position as rs and $r's'$. The strain and the stress are greatest at the outermost fibre and are zero at the center. At any intermediate point the strain and stress are proportional to the distance from the center. From this, as in flexure, it can readily be seen that the resistance of the section is not proportional to the area but to some other factor

Z , which is here found to be $\pi d^3 \div 16$. Notice that in Church Pars. 94 and 219 we find the moment of inertia of a shaft subjected to torsion to be $\pi r^4 \div 2$ and if this be divided by r , the distance

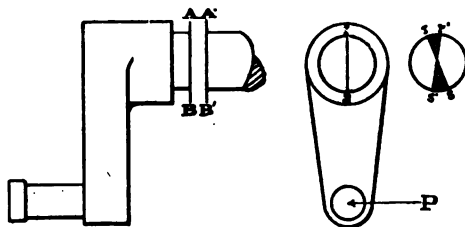


FIG. 23.

from the center to the outermost fibre, it gives $\pi r^3 \div 2$ or $\pi d^3 \div 16$ as stated above, where d is the diameter of the shaft.

Taking f as the maximum shearing stress and substituting in the formula $M = f Z$ we have

$$P R = f \frac{\pi d^3}{16} = .19635 d^3 f$$

$$d = 1.72 \sqrt[3]{\frac{P R}{f}} = 1.72 \sqrt[3]{\frac{T}{f}} \quad (4)$$

For a square shaft $M = .235 b^3 f$ where b is the side of the square.

For wrought iron shafts $f = 6000$ to 9000 .

For mild steel shaft $f = 9000$ to 12000 .

Formula (4) would not be used directly on shafts subjected to both bending and twisting, although, in modified form, it does apply. See Par. 68.

67. Hollow Shafts:—The value of Z for a hollow shaft subjected to a twisting moment is $\pi (d_1^4 - d_2^4) \div 16 d_1$ where d_1 is the outer diameter and d_2 is the inner diameter. This can then be substituted in $M = f Z$ as above. A very common requirement in machine design is the following:

Given a solid shaft d inches in diameter, it is desired to replace this with a hollow shaft of equal weight having d_1 inches external diameter. What is the internal diameter?; or the following: Given a solid shaft d inches diameter and capable of transmitting horse power under stated conditions, it is desired to replace this shaft with a light hollow shaft capable of resisting the same twisting moment and having d_1 inches external diameter: What will be the internal diameter d_2 ?

If the material is the same in each case we have

$$(1) \left\{ \frac{\text{Weight of hollow shaft}}{\text{Weight of solid shaft}} = \frac{d_1^2 - d_2^2}{d^2} \right. \quad (5)$$

Which gives $d^2 = d_1^2 - d_2^2$ for equal weights.

$$(2) \left\{ \frac{\text{Strength of hollow shaft}}{\text{Strength of solid shaft}} = \frac{d_1^4 - d_2^4}{d_1^4} \right. \quad (6)$$

Which gives $d^3 = \frac{d_1^4 - d_2^4}{d_1}$ for equal strengths.

Application:—A two-inch shaft is transmitting 33 H. P. at 250 R. P. M. Find (1) Maximum twisting moment; (2) maximum fibre stress; (3) internal diameter d_2 of a hollow shaft 2.5 inches external diameter that is capable of resisting the same twisting moment as the solid shaft.

$$(1) \text{ H. P.} = \frac{2 \pi R N P}{33000 \times 12} ; P. R. = 7570''^{\#} = M.$$

$$(2) f = \frac{M}{Z} ; Z = \frac{\pi d^3}{16} ; f = 4802''^{\#} [1]$$

$$(3) d^3 = \frac{d_1^4 - d_2^4}{d_1} ; d_2 = 2.087'' \text{ say } 2''.$$

If we wish a hollow shaft to be say 3 inches external diameter and of equal weight we have $d^2 = d_1^2 - d_2^2$; $d_2 = 2.25''$. A comparison of the above sizes is shown in Fig. 24.

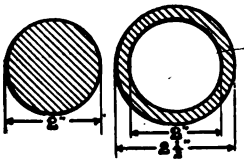
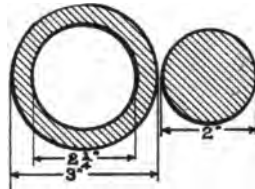


FIG. 24.



Equal Strength.

Equal Weight.

68. Combined Twisting and Bending:—This combined stress is present in many short shafts, as counter shafts, and in shafts carrying gears and pulleys, consequently the problem becomes

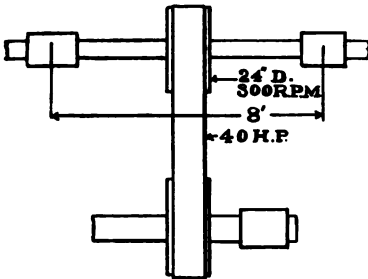


FIG. 25.

very important one for the engineer. Most shafting is subjected to a twisting and a bending stress simultaneously and would be classified under this head. Take for illustration a piece of shafting receiving power as in Fig. 25 with the thrust from the belt acting at the point where it will produce the greatest bending moment, i. e., midway between the bearings; we have a shearing stress in the fibres due to the twisting moment and

a bending stress due to the pull of the belt. These stresses may be combined into an *equivalent shearing stress* or into an *equivalent bending stress*. To find the diameter of such a shaft by the method employing the equivalent shearing stress, find the twisting moment T_m , (P. R.) as in Par. 65, and the bending moment B_m (in this case $Wl \div 4$), and substitute in formula (7) for the equivalent twisting moment T_e . After finding T_e substitute this value for T in formula (4). The reverse of this process, in many cases, may be employed.

Application.—From the H. P. formula

$$T_m = \text{P. R.} = \frac{H P \times 12 \times 33000}{2 \pi N} = 8403 \text{ ''\#}$$

$$\therefore P = \frac{8403}{12} = 700\#$$

Here the total side thrust on the shaft due to the belt is $W = 2 \times 700 = 1400\#$.

Note.—Assume the ratio $\frac{T_2}{T_1} = 3$ which may be considered good condition of service. See Par. 89. Tests by William Sellers & Co. This ratio may also be checked theoretically by formula (15), Par. 98. We then have from Fig. 22, $T_2 - T_1 = P$, and by combining obtain $T_2 = 3P \div 2$. Since the total side thrust from the belt is due to $T_2 + T_1$ we have

$$T_2 + T_1 = \frac{3P}{2} + \frac{P}{2} = 2P = W.$$

According to this the shaft will be supporting 1400 pounds at the center as given above and will have a bending moment of

$$B_m = \frac{Wl}{4} = \frac{1400 \times 12 \times 8}{4} = 33600 \text{ ''\#}$$

From Low and Bevis Par. 135 the formula is obtained for combined twisting and bending.

$$T_e = B_m + \sqrt{B_m^2 + T_m^2} \quad (7)$$

Where T_e = Equivalent twisting moment, which when found would be substituted for T in formula (4).

B_m = bending moment.

T_m = twisting moment.

Substituting in (7) the values of P. R. and $\frac{Wl}{4}$ we have

$$T_e = 33600 + \sqrt{(33600)^2 + (8403)^2} = 68230.$$

Substituting this value of T_e into formula (4) and taking the suggested value of f from the same reference as 9000 we have

$$d = 1.72 \sqrt[3]{\frac{68230}{9000}} = 3.37'' \text{ say } 3\frac{3}{8}''$$

The conditions just worked through would be very similar to those of a *jack shaft* or of a shaft serving the same duty as a jack shaft. The shaft is very large because of the unusual bending moment. If the bending moment be reduced by making the distance between the bearings less the diameter of the shaft would become less. Suppose this be made 6 feet instead of 8 feet, our formula would then become

$$T_e = 25200 + \sqrt{(25200)^2 + (8403)^2} = 51765''\#$$

$$d = 1.72 \sqrt[3]{\frac{51765}{9000}} = 3.07'' \text{ say } 3''$$

The following empirical formulas are frequently used for mild steel shafting:—

69. High Speed Engine Shaft:—Barr, Trans. A. S. M. E., Vol. 18, Page 756.

$$d = 7.3 \sqrt[3]{\frac{\text{H. P.}}{\text{R. P. M.}}} \quad (8)$$

70. Dynamo Shaft:—

$$d = K_1 \sqrt[4]{\frac{W}{\text{R. P. M.}}} \quad (9)$$

Where d = diameter of shaft in the armature core.

W = output in watts.

K_1 = constant depending on output as per table.

KW	1	5	10	50	100	200	500	1000	2000
K_1	1	1.1	1.2	1.3	1.4	1.5	1.6	1.7	1.8

71. Counter Shafts subjected to a heavy bending load:—

$$d = \sqrt[3]{\frac{125 \text{ H. P.}}{\text{R. P. M.}}} \quad (10)$$

72. Ordinary Line Shafting subjected to both bending and twisting:—

$$d = \sqrt[3]{\frac{90 \text{ H. P.}}{\text{R. P. M.}}} \quad (11)$$

73. Transmission Shafting in which the only bending stress is that due to its own weight:—

$$d = \sqrt[3]{\frac{62.5 \text{ H. P.}}{\text{R. P. M.}}} \quad (12)$$

74. Classification of Shafts and Shafting:—

Prime Movers....	{	Engine Shaft.
		Dynamo Shaft.
		Turbine Shaft.
Transmission Lines	{	Jack Shaft.
		Lines of Shafting.
		Counter Shaft.

Machine Shafts.

75. Materials Used in Shafting are wrought iron and mild steel. Steel shafting is more rigid, can be made more cheaply and is superseding wrought iron shafting to a large degree.

76. Process of Manufacture:—

Hot Rolled.—Dark rough surface, commonly called *black shafting*.

Cold Rolled.—Bright smooth surface.

77. Preparation for the Market:—

Hot Rolled.—Sometimes turned only at bearings and couplings; usually machine finish all over.

Cold Rolled.—Needs no finish other than that received by the die at the mills, should not be turned, and is very accurately sized for all diameters.

Turned shafting is made from standard hot-rolled rods by turning off 1-16 inch in diameter. The following table gives the standard sizes prepared for the market:

Size of Rod in inches.....	1	1½	1¼	1⅜	1½	1⅝	1¾	1⅞	2	2¼	2½	3
Size of Shafting in inches.....	1⅜	1⅞	1⅞	1⅞	1⅞	1⅞	1⅞	1⅞	1⅞	2⅞	2⅞	2⅞

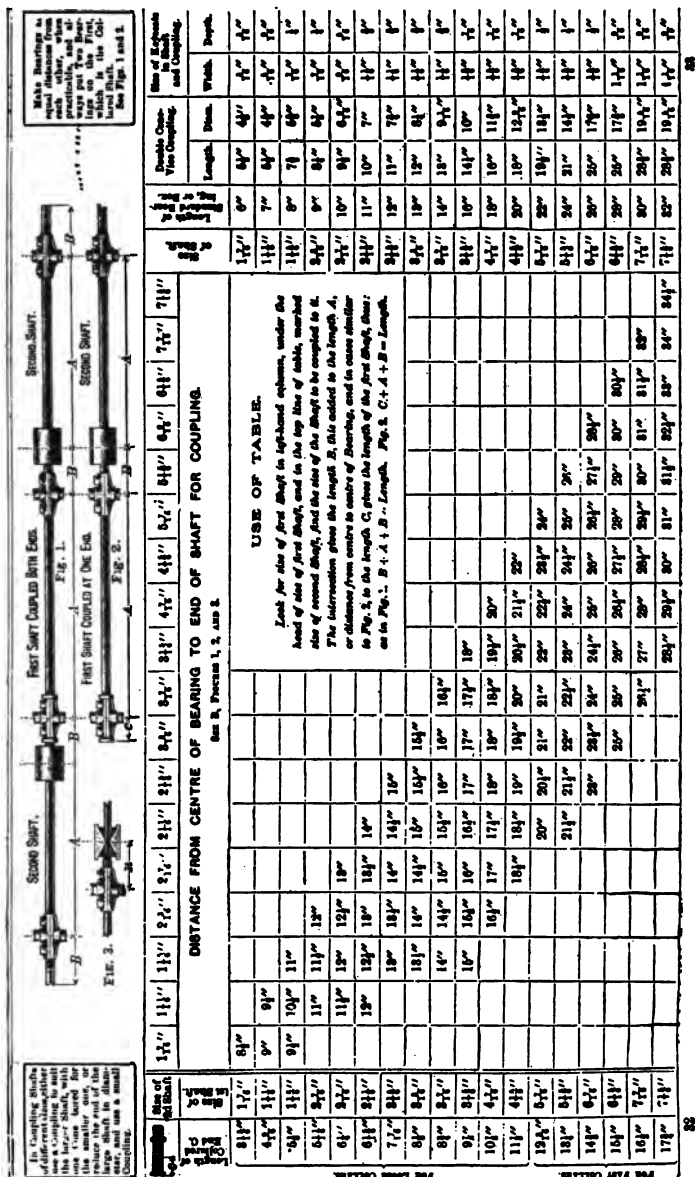
Cold rolled shafting is made by drawing it through a die the exact diameter of the finished shafting. Cold rolled shafting is about 25% better to resist twisting and bending than turned shafting of the same diameter and the same material.

78. Length of Shafts:—Shafting is manufactured in standard lengths but it can be ordered cut to almost any length up to twenty-five feet.

It is always wise in ordering shafting to specify the exact diameter, length, kind of material and finish; as *20 feet of 2 inch cold rolled steel shafting* or *20 feet of turned steel shafting 1 1/8 inches actual diameter*. The reason for this is because the turned shafting

is usually rated the same diameter as the rough rod from which it was turned; thus a piece of 2 inch turned shafting is $1\frac{7}{8}$ inches actual diameter.

TABLE VI—LAYING OUT SHAFTS.



Wherever it is possible to do so, pulleys should be set next to the journals to relieve the shaft from the bending due to the pull of the belt.

79. Speed of Shafting:—The best speed for shafting lines is 300 R. P. M. This may be reduced for iron working machinery to 200 R. P. M. and may be increased for wood working machinery to 400 R. P. M.

80. Speed of Counter Shafts:—
Iron Working Machines 80 to 150 R. P. M.
Wood Working Machines 200 to 1000 R. P. M.

81. Hangers and Shaft Supports:The following diagrams show some of the different methods of supporting shafts, and give data referring to these various forms.

TABLES VII. A. TO R.

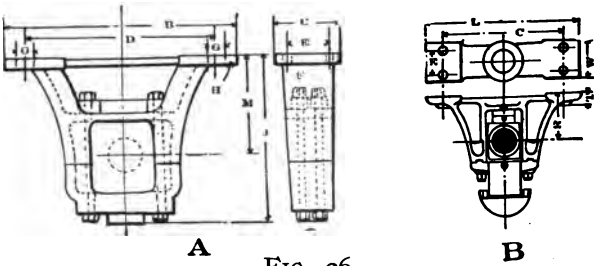


FIG. 26.

HEAD-SHAFT HANGERS.—A

Size of Shaft in inches	B Inch.	C Inch.	D Inch.	E Inch.	F 4 Bolts. D'am. Inch.	G Inch.	H Inch.	J Inch.	M Inch.
2¼-3	25	6	21½	3½	¾	2	1½	18½	12
3½-4	28	8	24	5	1	2¼	1½	19½	12
4¼-5	31	9	26	5½	1½	2½	1½	21	12
5¼-6	34	10	29	6½	1¾	2¾	1½	22	12
6¼-7	36½	11	31½	7½	1¾	3	1¾	23½	12
7¼-8	38	12	32½	7½	1¾	3	2	24½	12

HEAD-SHAFT HANGERS.—B

Size of Shaft	R Drop	L Length of Foot	W Width of Foot	T Thick-ness of Foot	C Centre to Centre of Bolts	E Centre to Centre of Bolts	Numb'r and Size of Bolts
3½"	11"	34"	7½"	1¼"	26"	4¾"	4-1"
4½"	11"	35½"	8½"	1½"	27½"	5"	4-1"
4¾"	11"	37½"	9½"	1¾"	28¾"	5½"	4-1"
5½"	11"	40½"	10¼"	2½"	31½"	6"	4-1½"

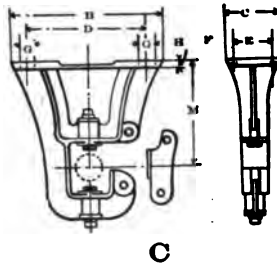


FIG. 27.

BALL AND SOCKET DROP HANGER FRAMES—C

Size of Shaft in inches	M	B	C	D	E	G	H	No. and Diam. of Bolts at F
2½ to 2½	10	17	5	13½	1½	¾	2—¾
	12	18½	5½	14½	1½	¾	2—¾
	14	18½	5½	14½	2½	1½	¾	4—¾
	16	19½	6	15½	3½	1½	¾	4—¾
	18	21½	6½	17½	3½	1½	¾	4—¾
	20	23½	6½	18½	3½	1½	¾	4—¾
	24	26½	7½	21½	4½	2½	1	4—¾
	30	26½	8½	20½	4½	2½	1	4—¾
	36	30½	8½	24½	4½	2½	1	4—¾
2½ to 3	10	16½	5½	12½	2½	1½	¾	4—¾
	12	17½	5½	13½	3	1½	¾	4—¾
	14	17½	6	13½	3½	1½	¾	4—¾
	16	19½	6½	15½	3½	1½	¾	4—¾
	18	21½	6½	17½	3½	1½	1	4—¾
	20	24½	7½	19½	4½	2½	1	4—¾
	24	28½	9	23½	5	2½	1½	4—¾
	30	29½	9½	23½	4½	2½	1½	4—¾
	36	32½	9½	26	4½	2½	1½	4—1
3½ to 3½	12	18	6½	14½	3½	1½	1	4—¾
	14	20	6½	15½	3½	1½	1	4—¾
	16	20½	6½	16½	3½	1½	1	4—¾
	18	23½	8½	18½	4½	2½	1½	4—¾
	20	27	8½	21½	5	2½	1½	4—¾
	24	31½	10	25½	6	2½	1½	4—¾
	30	30½	8½	25	5½	2½	1½	4—1
	36	35½	9½	30½	5½	2½	1½	4—1½

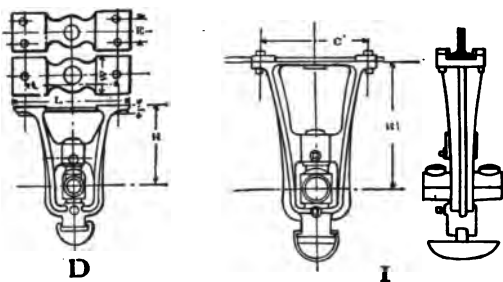


FIG. 28.

BALL AND SOCKET LINE HANGERS—D and I

1 $\frac{1}{16}$ " HANGER

R Drop	L Length of Foot	W Width of Foot	T Thickn'ss of Foot	C Centre to Centre of Bolts	E Centre to Centre of Bolts	No. and Size of Bolts
Inches	Inches	Inches	Inches	Inches	Inches	No. In.
8	13 $\frac{1}{4}$	5	$\frac{3}{4}$	10 $\frac{1}{4}$	2— $\frac{5}{8}$
10	13 $\frac{1}{2}$	5 $\frac{1}{2}$	$\frac{3}{4}$	10 $\frac{1}{2}$	2— $\frac{5}{8}$
11	14 $\frac{1}{4}$	5 $\frac{3}{4}$	$\frac{3}{4}$	11 $\frac{1}{4}$	2— $\frac{5}{8}$
13	14 $\frac{3}{4}$	6	$\frac{3}{4}$	11 $\frac{3}{4}$	2— $\frac{5}{8}$
16	17 $\frac{3}{4}$	6 $\frac{1}{4}$	$\frac{3}{4}$	14 $\frac{3}{4}$	2— $\frac{5}{8}$
18	18 $\frac{3}{4}$	6 $\frac{1}{2}$	$\frac{3}{4}$	15 $\frac{1}{4}$	3 $\frac{1}{4}$	4— $\frac{1}{2}$
20	20	6 $\frac{1}{2}$	$\frac{3}{4}$	16 $\frac{3}{4}$	3 $\frac{1}{2}$	4— $\frac{1}{2}$

1 $\frac{1}{8}$ " HANGER

R	L	W	T	C	E	No. and Size of Bolts
Inches	Inches	Inches	Inches	Inches	Inches	No. In.
8	14 $\frac{1}{4}$	5 $\frac{1}{2}$	$\frac{7}{8}$	10 $\frac{3}{4}$	2— $\frac{3}{4}$
10	15 $\frac{1}{4}$	6	$\frac{7}{8}$	11 $\frac{3}{4}$	2— $\frac{3}{4}$
11	16 $\frac{1}{4}$	6 $\frac{1}{4}$	$\frac{7}{8}$	12 $\frac{3}{4}$	2— $\frac{3}{4}$
13	17 $\frac{3}{4}$	6 $\frac{3}{4}$	$\frac{7}{8}$	13 $\frac{3}{4}$	2— $\frac{3}{4}$
16	19 $\frac{3}{4}$	7	$\frac{7}{8}$	15 $\frac{3}{4}$	2— $\frac{3}{4}$
18	21	7 $\frac{1}{4}$	$\frac{7}{8}$	17	4	4— $\frac{5}{8}$
20	22 $\frac{1}{4}$	7 $\frac{1}{2}$	$\frac{7}{8}$	18 $\frac{1}{4}$	4 $\frac{1}{2}$	4— $\frac{5}{8}$
25	25 $\frac{1}{4}$	9	$\frac{7}{8}$	21 $\frac{1}{4}$	5 $\frac{1}{4}$	4— $\frac{5}{8}$
30	27 $\frac{1}{4}$	10	$\frac{7}{8}$	23 $\frac{1}{4}$	6	4— $\frac{5}{8}$

SHAFTING SUPPORTS.

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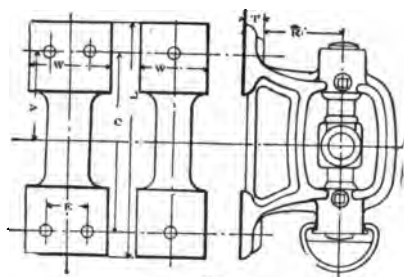
2½' HANGER						
8	17	6	1	12	2-½
10	18½	6½	1	13½	2-½
11	18½	6½	1	13½	2-½
13	20½	7	1	15½	2-½
16	22½	7½	1	16½	2-½
18	24	7½	1	18½	4	4-½
20	25	8	1	20	4½	4-½
25	27½	9½	1	2½	5½	4-½
30	30½	10½	1	25½	6½	4-½

2½' HANGER						
8	19	6½	1½	13	2-1
10	20½	6½	1½	14½	2-1
11	21½	7½	1½	15½	2-1
13	22	7½	1½	16	2-1
16	24½	8	1½	18½	2-1
18	25½	8½	1½	19½	4½	4-½
20	26½	8½	1½	20½	5	4-½
25	29½	10	1½	23½	6	4-½
30	32½	11	1½	26½	6½	4-½
36	36½	12	1½	30½	7½	4-½

3½' HANGER						
11	21½	7½	1¾	14½	2-1
11	23½	7½	1¾	16½	2-1
13	25½	8	1¾	17½	2-1
16	26½	8½	1¾	19½	2-1
18	27	8½	1¾	20	5	4-½
20	28½	9	1¾	21½	5½	4-½
25	31½	10½	1¾	24½	6½	4-½
30	34½	11½	1¾	27½	6½	4-½
36	37½	12½	1¾	30½	7½	4-½

3½' HANGER						
17	25½	8½	1½	17½	2-1½
11	25½	8½	1½	17½	2-1½
13	26½	8½	1½	18½	2-1½
16	28½	9	1½	20½	2-1½
18	29½	9½	1½	21½	5	4-1
20	30½	9½	1½	22½	6	4-1
25	33½	11	1½	25½	6½	4-1
30	37½	12	1½	29½	7	4-1
36	40½	13	1½	3½	7½	4-1

4½' HANGER						
11	29½	9½	1½	19½	5½	4-1½
13	30	10	1½	20	5½	4-1½
16	32	10½	1½	22	6½	4-1½
18	34½	10½	1½	24½	6½	4-1½
20	35½	10½	1½	25½	6½	4-1½
25	38½	12	1½	28½	7½	4-1½



E

FIG. 29.

DIMENSIONS OF BALL- AND-SOCKET POST HANGERS—E
For Ring Self-Oiling Boxes

Size of Shaft	L Length of Foot	W Width of Foot	T Thick-ness of Foot	C Centre to Centre of Bolt Holes	E Centre to Centre of Bolt Holes Horizontal	V Upper Bolt Hole above Centre of Shaft	R Foot to Centre of Shaft	Diam-eter of Bolts
Inch's	Inch's	Inch's	Inches	Inches	Incher	Inches	Inches	Inches
1 7/8	12	3 3/4	1 1/4	9	4 3/4	6	5/8
1 1/2	13 3/4	3 3/4	1 1/4	9 3/4	5 1/4	6	5/8
1 1/4	15	4	1 1/4	11	5 3/4	6	5/8
2 3/8	16 1/2	4 1/4	1 1/4	12	6 3/4	6	3/4
2 1/2	18	5	1 1/4	13	6 3/4	6	3/4
2 1/4	19	5 1/4	1 1/4	13 1/4	7 1/4	6	3/4
2 1/8	20	6	1 1/4	14	7 3/4	6	1
3 3/8	22	6 1/4	1 1/4	15 1/4	8 3/4	6	1 1/8
3 1/2	20 1/4	9 1/4	1 1/4	14	5 1/4	7 3/4	6	3/4
3 1/4	22 1/4	9 3/4	1 1/4	15 1/4	6	8 3/4	6	1
4 1/8	24 1/4	10	1 1/4	16 3/4	6 3/4	10	7 1/4	1 1/4
4 1/4	25 3/4	10 1/4	1 1/4	17 3/4	7 1/4	10 3/4	7 1/4	1 1/4

FOR STANDARD BOXES

Size of Shaft	L Length of Foot	W Width of Foot	T Thick-ness of Foot	C Centre to Centre of Bolt Holes	E Centre to Centre of Bolt Holes Horizontal	V Upper Bolt Hole above Centre of Shaft	R Foot to Centre of Shaft	Diam-eter of Bolts
Inch's	Inch's	Inch's	Inches	Inches	Incher	Inches	Inches	Inches
1 7/8	12	3 3/4	1 1/4	9	4 3/4	6	5/8
1 1/2	13 3/4	3 3/4	1 1/4	9 3/4	5 1/4	6	5/8
1 1/4	15	4	1 1/4	11	5 3/4	6	5/8
2 3/8	16 1/2	4 1/4	1 1/4	12	6 3/4	6	3/4
2 1/2	18	5	1 1/4	13	6 3/4	6	3/4
2 1/4	19	5 1/4	1 1/4	13 1/4	7 1/4	6	3/4
2 1/8	20	6	1 1/4	14	7 3/4	6	1
3 3/8	22	6 1/4	1 1/4	15 1/4	8 3/4	6	1 1/8
3 1/2	20 1/4	9 1/4	1 1/4	14	5 1/4	7 3/4	6	3/4
3 1/4	22 1/4	9 3/4	1 1/4	15 1/4	6	8 3/4	6	1
4 1/8	24 1/4	10	1 1/4	16 3/4	6 3/4	10	6	1 1/4
4 1/4	25 3/4	10 1/4	1 1/4	17 3/4	7 1/4	10 3/4	6	1 1/4

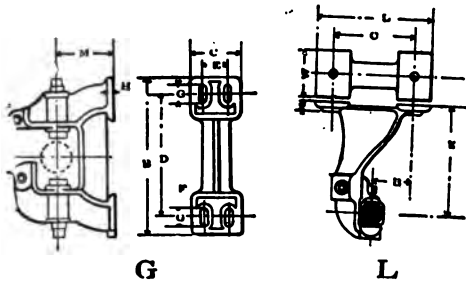


FIG. 30.

WALL BRACKET HANGERS—G

Size of Shaft, inches	B	C	D	E	No. and Size of Bolts, F	G	H	M
1 to 1½	16	4½	13½	2—½	1½	¾	5½
1½ to 2	18½	4½	15½	2—½	1½	¾	5½
2 to 2½	19½	5½	15½	2½	4—½	1½	¾	6½
2½ to 3	20½	6½	16½	3½	4—½	1½	¾	6½
3 to 3½	21½	7½	17½	4½	4—½	2	1	8
3½ to 4	21½	7½	17½	3½	4—½	2	1	8½
4 to 4½	25½	9	20½	5½	4—½	2	1½	8½

DIMENSIONS OF COUNTER HANGERS—L

With or Without Belt Shifter Arms.

Size of Shaft	R	L	W	T	C	E	B	No. and Size of Bolts
Inch.	Inch.	Inch.	Inch.	Inches	Inches	Inches	Inches	Inches
1½	8	10½	4¾	¾	6¾	2¾	2—¾
	10	11½	5	¾	7¾	3½	2—¾
	13	12¾	5½	¾	8¾	3½	2—¾
	16	14½	5½	¾	11½	4½	2—¾
	20	17½	6½	¾	13½	3½	6½	4—½
	25	19½	6½	¾	16½	3½	8½	4—½
1¾	13	12¾	5½	¾	9¾	3¾	2—¾
	16	14½	5½	¾	11¾	4¾	2—¾
1½	8	10½	4¾	¾	7¾	1½	2—¾
	11	11½	5½	¾	8	3	2—¾
	13	13½	5½	¾	10	4¾	2—¾
	16	15½	5½	¾	11¾	4¾	2—¾
	20	17½	6½	¾	14½	3¾	7	4—¾
	25	20½	7	¾	17½	4	9½	4—¾
1½	13	13¾	5½	¾	9¾	3½	2—¾
	16	15½	6½	¾	11¾	4¾	2—¾
	20	18½	7	¾	14½	3¾	7½	4—¾
	25	21	7½	¾	17	4	7½	4—¾

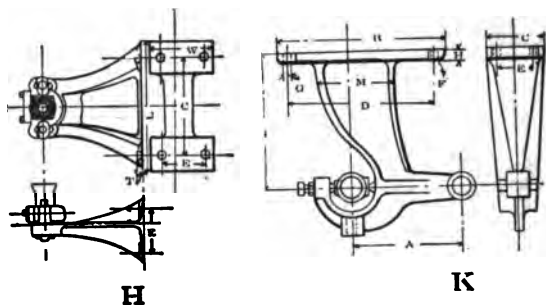


FIG. 31.

DIMENSIONS OF WALL HANGER BRACKETS—H

No. of Bracket	Size of Shaft	L	W	T	C	E	F	G	Size of Bolts	Projection from Wall
	Inch.	In.	In.	In.	In.	In.	In.	In.	In.	Inches
20	1 7/8 1 1/2 1 1/8	18	9	3/4	13 1/2	6	2	1 1/8 1 1/4 1 1/2	3/8 1/2 5/8	{ a=13 1/4 to 15 1/4 b=15 1/2 to 17 1/2 c=17 1/4 to 19 1/4
21	1 7/8 1 1/2 1 1/8	22	10	3/4	17 1/2	7	2 3/8	1 1/8 1 1/4 1 1/2	3/8 1/2 5/8	{ d=19 1/4 to 21 1/4 e=21 1/4 to 23 1/4 f=23 1/4 to 25 1/4
22	1 7/8 1 1/2 1 1/8	25	11	3/4	20 1/2	8	2 1/2	1 1/8 1 1/4 1 1/2	3/8 1/2 5/8	{ g=25 1/4 to 27 h=27 to 29 i=29 to 31
23	2 3/8 2 1/4 2 1/8	22	9 1/2	1 1/4	17	7	2	2 1/4 2 1/8 2 1/2	3/8 1/2 5/8	{ a=13 1/4 to 15 1/4 b=15 1/2 to 17 1/2 c=17 1/4 to 19 1/4
24	2 3/8 2 1/4 2 1/8	25	10 1/2	1 1/4	20	7 3/4	2 3/8	2 1/4 2 1/8 2 1/2	3/8 1/2 5/8	{ d=19 1/4 to 21 1/4 e=21 1/4 to 23 1/4 f=23 1/4 to 25 1/4
25	2 3/8 2 1/4 2 1/8	28	11 1/2	1 1/4	23	8 3/8	2 3/8	2 1/4 2 1/8 2 1/2	3/8 1/2 5/8	{ g=25 1/4 to 27 h=27 to 29 i=29 to 31
26	2 3/8 2 1/4 2 1/8	25	10 1/2	1 1/4	19	7 1/2	2 1/4	2 1/8 2 1/4 2 1/2	3/8 1/2 5/8	{ a=13 1/4 to 15 1/4 b=15 1/2 to 17 1/2 c=17 1/4 to 19 1/4

COUNTER HANGERS—K

Size of Shaft in Inches	B	C	D	E	F 4 Bolts D'am.	G	H	J	M
	Inch.	Inch.	Inch.	Inch.	Inch.	Inch.	Inch.	Inch.	Inch.
1 1/2—1 3/4	10 1/4	4 1/2	7 3/4	3 3/4	1/2	1 3/4	1	7 1/4	3 3/4
2	11	5 1/2	8 3/4	4 1/4	3/4	1 3/4	1 1/8	8 1/4	4
2 1/4—2 1/2	12 3/4	6 1/2	9 3/4	4 3/4	3/4	1 3/4	1 1/8	9 1/4	4 1/4
2 1/2—3	14 1/4	7 1/4	10 3/4	5 1/4	3/4	2	1 3/8	11 1/4	5 1/4
3 1/4—3 1/2	16 1/4	7 3/4	12 1/4	6	3/4	2 1/4	1 1/2	12 1/4	6
3 1/2—4	19 3/4	9	14 1/4	6 3/4	3/4	3 1/4	1 3/8	13 3/4	6 1/2

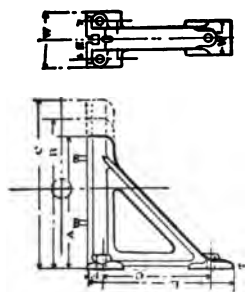


FIG. 32.

DIMENSIONS OF WALL BRACKETS FOR PILLOW BLOCKS—J

Number of Bracket	L Length of Base Inches	W Width of Upper Foot Inches	W' Width of Lower Foot Inches	T Thickness of Feet Inches	F Distance from Top of Bracket to Upper Bolt Holes Inches	E Distance between Upper Bolt Holes Inches	C Distance between Upper and Lower Bolt Holes Inches	Projection from Wall			Size of Bolt Inches
								a Least Inches	b Mean Inches	c Greatest Inches	
1	14	6	3	1½	1½	4	11	13	15	17	¾
2	15	6	3	1½	1½	4	12	19	21	23	¾
3	18	6½	3	1½	1½	4½	15	25	27	29	¾
4	20½	7	4	1½	1½	5	17½	31	33	35½	¾
5	16½	6½	4	1½	1½	4½	13	15½	18½	21½	¾
6	19½	7½	4	1½	1½	5	16	23½	26½	29½	¾
7	25	7½	5	1½	1½	5½	21½	31½	34½	37½	¾
8	22	8	5	1½	2	6	17½	19½	24½	28½	¾
9	28½	9½	5½	1½	2	7	23½	31½	34½	38	¾
10	23½	9½	7	2	2½	6½	19½	23	27	30½	1
11	31½	11	7	2	3	7½	25½	33	36½	40½	1
12	28½	11	7	2½	3	7½	21	26	30	34	1½
13	36	13	9	2½	3	9	28½	36½	39½	42½	1½

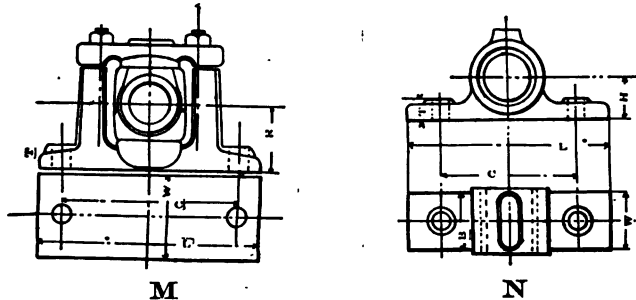


FIG. 33.

DIMENSIONS OF PILLOW BLOCKS—M
With Standard Boxes

Size of Shaft	L Length of Foot	W Width of Foot	T Thick- ness of Foot	C Centre to Centre of Bolt Holes	R Base to Centre of Shaft	Number and Diameter of Bolts
inches	Inches	Inches	Inches	Inches	Inches	Inches
1½	6½	2½	1½	5	1½	2—¼
1¾	7¼	2¾	1¾	5¾	2	2—¼
1⅞	7¾	3	1¾	6¼	2½	2—¼
1⅞	8	3¼	1¾	7¼	2½	2—¼
1⅞	8½	3½	1¾	8	2¾	2—¾
1⅞	10¼	3¾	1¾	8½	3½	2—¾
2	10½	4¼	1¾	9	3¾	2—¾
2¼	11½	4¾	1¾	10	3¾	2—¾
2½	12½	5	1¾	10¾	4	2—¾
2½	13½	5½	1¾	11½	4½	2—¾
3	14	5¾	1¾	12½	4¾	2—1
3¼	16	6¾	2	14	5½	2—1
3½	18	7½	2	16¾	6½	2—1½
4	20¾	8¾	2½	17¼	6½	2—1½
4½	21½	9½	2½	19¼	7¾	2—1½
5	24	10	2½	20½	8	2—1½
5½	25½	10¾	2½			

SOLID PILLOW BLOCK—N

	Inches	Inches	Inches	Inches	Inch.	Inch.
Size of Bore.....	1½	2½	2½	2½	3½	3½
B. Length of bearing .	6	6¼	7¼	9	10¼	12
L. Length of foot	8¼	9¼	10¼	11¾	13¼	1½
W. Width of foot	3½	4	4½	5½	6½	7½
T. Thickness of foot . . .	1	1½	1½	1½	1½	2
H. Height to centre . . .	2	2¼	2½	3	3½	4
C. Between bolt centres	6¼	8	8½	9	11	12
Size of bolts	¾	¾	¾	¾	1	1

SHAFTING SUPPORTS.

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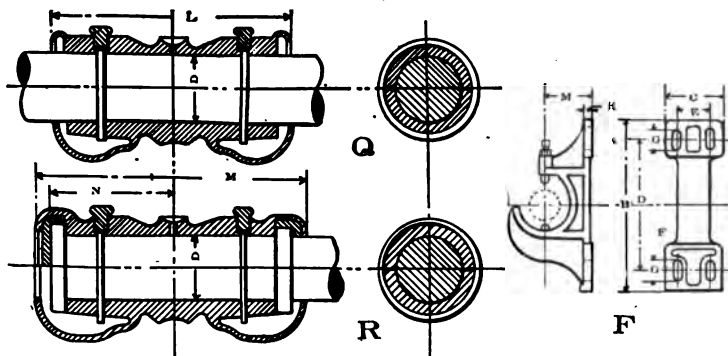


FIG. 34.

SELF OILING BEARINGS—QR
With or Without Collars

Size of Shaft Inches	3 Diam. Box		4 Diam. Box	
	M With Collars	L Without Collars	M With Collars	L Without Collars
	Inches	Inches	Inches	Inches
1 1/8			9 1/8	7 1/4
1 1/4	8 1/2	6 7/8	10 1/4	8 1/8
1 1/2	9 3/8	7 3/8	11 1/8	9 3/8
1 3/4	10 1/2	8 1/8	12 1/8	10 7/8
2 1/8	11 1/8	9 1/8	13 1/8	11 1/2
2 1/4	12 1/8	10 1/8	14 1/8	12 1/8
2 1/2	13 1/8	11 1/8	15 1/8	13 1/8
2 3/4	14 1/2	12 1/8	16 1/8	14 1/8
3 1/8	15 1/8	13 1/8	17 1/8	15 1/8
3 1/4	16 1/8	14 1/8	18 1/8	16 1/8
3 1/2	17 1/8	15 1/8	19 1/8	17 1/8
3 3/4	18 1/8	16 1/8	20 1/8	18 1/8
4 1/8	19 1/8	17 1/8	21 1/8	19 1/8
4 1/4	20 1/8	18 1/8	22 1/8	20 1/8
4 1/2	21 1/8	19 1/8	23 1/8	21 1/8
4 3/4	22 1/8	20 1/8	24 1/8	22 1/8
5 1/8	23 1/8	21 1/8	25 1/8	23 1/8
5 1/4	24 1/8	22 1/8	26 1/8	24 1/8
5 1/2	25 1/8	23 1/8	27 1/8	25 1/8
5 3/4	26 1/8	24 1/8	28 1/8	26 1/8
6 1/8	27 1/8	25 1/8	29 1/8	27 1/8
6 1/4	28 1/8	26 1/8	30 1/8	28 1/8
6 1/2	29 1/8	27 1/8	31 1/8	29 1/8
6 3/4	30 1/8	28 1/8	32 1/8	30 1/8
7 1/8	31 1/8	29 1/8	33 1/8	31 1/8
7 1/4	32 1/8	30 1/8	34 1/8	32 1/8
7 1/2	33 1/8	31 1/8	35 1/8	33 1/8
7 3/4	34 1/8	32 1/8	36 1/8	34 1/8
8 1/8	35 1/8	33 1/8	37 1/8	35 1/8
8 1/4	36 1/8	34 1/8	38 1/8	36 1/8
8 1/2	37 1/8	35 1/8	39 1/8	37 1/8
8 3/4	38 1/8	36 1/8	40 1/8	38 1/8
9 1/8	39 1/8	37 1/8	41 1/8	39 1/8
9 1/4	40 1/8	38 1/8	42 1/8	40 1/8
9 1/2	41 1/8	39 1/8	43 1/8	41 1/8
9 3/4	42 1/8	40 1/8	44 1/8	42 1/8
10 1/8	43 1/8	41 1/8	45 1/8	43 1/8
10 1/4	44 1/8	42 1/8	46 1/8	44 1/8
10 1/2	45 1/8	43 1/8	47 1/8	45 1/8
10 3/4	46 1/8	44 1/8	48 1/8	46 1/8
11 1/8	47 1/8	45 1/8	49 1/8	47 1/8
11 1/4	48 1/8	46 1/8	50 1/8	48 1/8
11 1/2	49 1/8	47 1/8	51 1/8	49 1/8
11 3/4	50 1/8	48 1/8	52 1/8	50 1/8
12 1/8	51 1/8	49 1/8	53 1/8	51 1/8
12 1/4	52 1/8	50 1/8	54 1/8	52 1/8
12 1/2	53 1/8	51 1/8	55 1/8	53 1/8
12 3/4	54 1/8	52 1/8	56 1/8	54 1/8
13 1/8	55 1/8	53 1/8	57 1/8	55 1/8
13 1/4	56 1/8	54 1/8	58 1/8	56 1/8
13 1/2	57 1/8	55 1/8	59 1/8	57 1/8
13 3/4	58 1/8	56 1/8	60 1/8	58 1/8
14 1/8	59 1/8	57 1/8	61 1/8	59 1/8
14 1/4	60 1/8	58 1/8	62 1/8	60 1/8
14 1/2	61 1/8	59 1/8	63 1/8	61 1/8
14 3/4	62 1/8	60 1/8	64 1/8	62 1/8
15 1/8	63 1/8	61 1/8	65 1/8	63 1/8
15 1/4	64 1/8	62 1/8	66 1/8	64 1/8
15 1/2	65 1/8	63 1/8	67 1/8	65 1/8
15 3/4	66 1/8	64 1/8	68 1/8	66 1/8
16 1/8	67 1/8	65 1/8	69 1/8	67 1/8
16 1/4	68 1/8	66 1/8	70 1/8	68 1/8
16 1/2	69 1/8	67 1/8	71 1/8	69 1/8
16 3/4	70 1/8	68 1/8	72 1/8	70 1/8
17 1/8	71 1/8	69 1/8	73 1/8	71 1/8
17 1/4	72 1/8	70 1/8	74 1/8	72 1/8
17 1/2	73 1/8	71 1/8	75 1/8	73 1/8
17 3/4	74 1/8	72 1/8	76 1/8	74 1/8
18 1/8	75 1/8	73 1/8	77 1/8	75 1/8
18 1/4	76 1/8	74 1/8	78 1/8	76 1/8
18 1/2	77 1/8	75 1/8	79 1/8	77 1/8
18 3/4	78 1/8	76 1/8	80 1/8	78 1/8
19 1/8	79 1/8	77 1/8	81 1/8	79 1/8
19 1/4	80 1/8	78 1/8	82 1/8	80 1/8
19 1/2	81 1/8	79 1/8	83 1/8	81 1/8
19 3/4	82 1/8	80 1/8	84 1/8	82 1/8
20 1/8	83 1/8	81 1/8	85 1/8	83 1/8
20 1/4	84 1/8	82 1/8	86 1/8	84 1/8
20 1/2	85 1/8	83 1/8	87 1/8	85 1/8
20 3/4	86 1/8	84 1/8	88 1/8	86 1/8
21 1/8	87 1/8	85 1/8	89 1/8	87 1/8
21 1/4	88 1/8	86 1/8	90 1/8	88 1/8
21 1/2	89 1/8	87 1/8	91 1/8	89 1/8
21 3/4	90 1/8	88 1/8	92 1/8	90 1/8
22 1/8	91 1/8	89 1/8	93 1/8	91 1/8
22 1/4	92 1/8	90 1/8	94 1/8	92 1/8
22 1/2	93 1/8	91 1/8	95 1/8	93 1/8
22 3/4	94 1/8	92 1/8	96 1/8	94 1/8
23 1/8	95 1/8	93 1/8	97 1/8	95 1/8
23 1/4	96 1/8	94 1/8	98 1/8	96 1/8
23 1/2	97 1/8	95 1/8	99 1/8	97 1/8
23 3/4	98 1/8	96 1/8	100 1/8	98 1/8
24 1/8	99 1/8	97 1/8		
24 1/4	100 1/8	98 1/8		
24 1/2		99 1/8		
24 3/4		100 1/8		

WALL BRACKET HANGERS—F

Size of Shaft in inches	B	C	D	E	No. and Size of Bolts. F	G	H	M
1 1/8 to 1 1/4	13 3/4	4	11 1/8	2—3/8	1 3/8	3/4	2 3/8
1 1/4 to 1 1/2	14 1/4	4 3/8	11 3/8	2—3/8	1 1/2	7/8	2 1/2
1 1/2 to 1 3/4	14 3/4	4 1/2	12	2—3/8	1 3/4	1	3
1 3/4 to 2 1/8	15 1/2	4 3/4	12 3/8	2—3/8	1 5/8	1	3 1/8
2 1/8 to 2 1/4	16 1/8	5	13 1/8	2—3/8	1 3/4	1	3 3/8
2 1/4 to 2 1/2	17 1/8	5 1/8	14 1/8	2—3/8	1 7/8	1	3 5/8
2 1/2 and 3	17 3/8	5 1/4	14	2—3/8	2	1	4 1/8
3 and 3 1/8	17 3/4	6	13 3/8	3 1/2	4—3/4	2 1/8	1	4 3/8
3 1/8 and 3 1/4	19 1/8	6 3/8	15	3 3/4	4—7/8	2 1/2	1	5 1/8
3 1/4 to 3 1/2	20 1/8	6 3/4	15 1/8	3 7/8	4—7/8	2 3/8	1	5 1/2
3 1/2 and 4	21	7	16 1/8	4	4—7/8	2 1/2	1	5 3/8

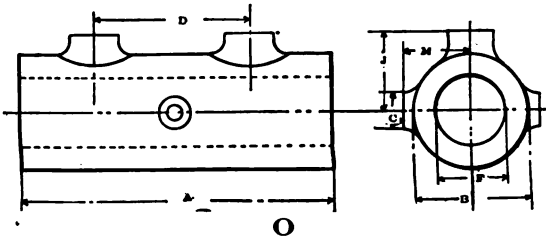


FIG. 35.

SOLID JOURNAL BOX—O

F Size of Shaft in Inches	A Inches	B Inches	C Inches	D Inches
1/8	4	7 1/4	2 1/2	5 1/4
1/4	4 1/4	7 3/4	3	5 5/8
1/2	4 1/2	9	3	6 1/4
3/4	5 1/4	10	3 1/2	7 1/4
1	6 1/4	10 1/2	3 1/2	7 3/4
1 1/8	6 3/4	11	3 3/4	8
1 1/4	7 1/4	11 1/4	4	8 1/2
1 1/2	8 1/4	12 1/4	4 1/4	9 1/4
1 3/4	8 3/4	12 3/4	4 3/4	9 3/4
2	9 1/4	13	5	10
2 1/4	11	13 1/4	5 1/4	11
2 1/2	11 1/4	15	5 5/8	11 1/4
2 3/4	12 1/4	15 1/4	6	11 3/4

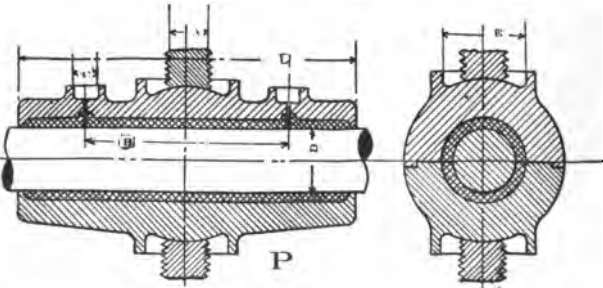


FIG. 36.

SPLIT JOURNAL BOX—P

D Size of Shaft in Inches	A	B	C	L	E
1½	¾	3¾	¼	5¼	2¼
1¾	¾	4	¾	6¼	2½
1¾	1½	4½	¾	6¾	2¾
1¾	1½	4¾	¾	7¼	2¾
1¾	1½	5¼	1½	7¾	3
1¾	1¾	5¾	1½	8¼	3¼
2	1¾	6¼	1½	9	3¾
2	1¾	7	¾	10¼	3¾
2	1¾	7½	¾	10¾	3¾
2	1¾	8¼	¾	10¾	4¼
3	2	9¼	1½	12	4½
3	2	10¼	1½	13	5
3	2	11¼	1½	13¾	5½
4	2½	13¼	1½	15¾	6
4	2½	14¼	1½	16¾	6½
5	3	16	1½	18¾	7
5	3	17¼	1½	19¾	7½

82. Journals:—The average distance from center to center of journals, on lines of wrought iron or steel shafting carrying belts and pulleys, varies from 10 feet on a 2 inch line to 18 feet on a 5 inch line. The maximum values are

2"	—	12'
3"	—	15'
4"	—	18'
5"	—	22'

83. Length of Journal:—The length of the journal is taken from 3d to 4d. For machine shafts this may under certain conditions be reduced to d. A good average for machine shafts is 2d.

84. Journal Box:—Journal boxes are made both solid and split. Arrangements for oiling are made as follows: first, by gravity feed oil cups; second, through a wick which dips down into an oil well in the box and laps over the top of the shaft; third, by a ring or chain which spans the shaft and has its lower part immersed in the oil in the well. When the shaft revolves this ring or chain moves with it thus carrying oil to the top of the shaft. The second and third are called *self oiling boxes*. Counter shafts usually have solid boxes with gravity feed oil cups. Line shafts usually have split boxes arranged for either wick, chain or ring oiling.

85. References for Shafting:—Reuleaux, "The Constructor," pages 92-95; Low and Bevis, "Machine Design," pages 94-99, 115-133; Church, "Mechanics of Engineering," pages 233-243.

86. Problems:—

(1) Eight hundred pounds of material are drawn from a depth of 90 feet by a rope weighing 1.15 pounds per linear foot; how many

units of work are expended? What H. P. would be required to raise the material in $4\frac{1}{2}$ minutes?

(2) A dynamometer-car between a locomotive exerting 500 H. P. and a train showed the draw-bar pull to be 3500 pounds. Find the speed of the train in miles per hour.

(3) A punching machine is so arranged that 40 holes, $\frac{3}{4}$ inches in diameter, can be punched in 3 minutes through a mild steel plate $\frac{1}{2}$ inch thick. Assuming 65,000 pounds per square inch to be the ultimate shearing strength of this piece of mild steel and that the average pressure is one-half the maximum, find how much work is performed by the machine in 1 minute, and the H. P. required to drive the machine, assuming 30 per cent friction loss.

(4) The travel of the table of a planer cutting both ways is 6 feet, and the resistance to be overcome while cutting is taken at 400 lbs. If the number of double strokes made in one hour be 70, find the H. P. absorbed by the machine.

(5) A shaft transmits 30 H. P. at 120 revolutions per minute. Find the twisting moment on the shaft. How many revolutions per minute must it make to transmit 55 H. P.?

(6) A shaft 3 inches in diameter transmits 20 H. P. How many H. P. will a shaft $4\frac{1}{2}$ inches in diameter transmit at half the number of revolutions of the former, and how many at twice that number?

(7) A steam engine 18 inches by 20 inches runs at 100 revolutions per minute and develops 150 H. P. Find twisting moment on crank shaft. A high speed engine of the same dimensions and with the same mean effective pressure runs at 250 revolutions per minute. Find the diameter of the crank shaft.

(8) One hundred H. P. is to be transmitted at 300 revolutions per minute. Find, (a), the diameter of an ordinary line shaft; (b), the diameter of a transmission shaft; and (c), the diameter of a counter shaft to meet the conditions.

(9) Find the H. P. that can be transmitted by a wrought iron shaft 4 inches in diameter, driven by a spur gear wheel 2 feet pitch diameter, at a speed of 120 revolutions per minute. Assume ultimate strength of 70,000 lbs. per sq. in. and factor of safety of 10.

(10) A mild steel shaft transmits 20 H. P. at 100 revolutions per minute. How many H. P. will it transmit at 180 revolutions per minute? Find the diameter of the shaft when 50 H. P. is transmitted at 150 revolutions per minute. Ultimate strength of mild steel 80,000 lbs. per square inch, factor of safety 10.

(11) The drum of a crane is 15 inches in diameter; the pull on the wire rope, which is 1 inch in diameter, is 4 tons. Find the diameter of the drum shaft. Safe working stress 6000 pounds per square inch.

12) Compare the torsional strengths and weights of two shafts of same material and of equal lengths one being 10 inches diameter

and solid, the other 10 inches diameter with a 6 inch hole through it.

(13) Twenty-five H. P. is transmitted by a belt passing over a pulley 30 inches in diameter located on a mild steel shaft midway between bearings set 10 feet center to center. If the shaft is running at 250 revolutions per minute with $T_2 = 3 T_1$ find the diameter of the shaft required.

Belt Transmission.

87. Belt Transmission Materials:—Belting is classified as to material, as leather, canvas and rubber. Leather is the standard belting for shop and power plant service. It should be kept dry and should not be used in high temperatures. Leather belting is made of strips of leather, which have been cut from the hides and joined together by glue or cement, and rivets. To make a thick belt, one or more of these layers are glued together hence the classification single, double, etc. For ultimate strength of leather belting see Par. 89.

Canvas belting is made from a number of layers of canvas stitched together and finally *sized*. The thickness agrees fairly well with that of leather belting. It can be used in damp places if necessary. Canvas belting has an ultimate tensile strength of about 5800 pounds per square inch. It is used largely on conveying machinery.

Rubber belting gives better contact to the pulley than leather or canvas, and is not affected by moisture. It is fairly uniform in width and thickness, will stand wide variation in heat and cold, and has a tensile strength of about 3500 pounds per square inch.

The under line of a horizontal belt should be the working line, in which case the sag of the upper line increases the arc of contact.

Single thickness belts should not be used much over 12 inches in width.

The *efficiency* of a belt is affected by the direction in which it operates; a vertical belt being the least efficient and a horizontal belt the most efficient. It is also affected by the velocity and the tension.

88. Velocity:—Belting is said to give out *its maximum efficiency* at 6000 feet per minute. (See Par. 102). The best current practice for high speed belts, however, is from 4000 to 4500 feet per minute. With a given velocity of belt and a known horse power, we can assume a working stress per square inch of section and solve for the square inches of belt area.

89. Fibre Stress:—Concerning the *working strength* of leather belting many references might be quoted, chiefly those in Kent, Pages 876-887. These references however, show such a lack of uniformity that it becomes largely a matter of the judgment of the designer. Tests of leather belting show an ultimate strength of

about 4000 pounds per square inch, which, with a factor of safety of 10, gives $f = 400$ pounds per square inch as a maximum working stress. This figure varies in practice from 250 to 400 pounds and agrees with a *turning force*, p , per square inch of section, between 180 and 290 pounds. Kent seems to favor a *turning force* of 275 pounds per square inch, which is equivalent in a single belt to 52 pounds and in a double belt to 86 pounds *per inch of width*. These figures seem to give results that agree well with current practice in high speed *cemented or endless belts*.

Taylor, Trans. A. S. M. E. Vol. XV, recommends the following values as representing the most economical practice for low speed *shop belts*, claiming that the "ordinary rules" place belting under entirely too severe total load for economy.

Average total load on belting, 200 to 225 pounds, per square inch section of belt.

Six-ply and seven-ply rubber belts, and double leather belts will transmit economically a pull of 30 to 35 pounds per inch of width.

Experience in actual service shows these values to be far more economical and satisfactory than those based on a lower factor of safety; particularly with regard to interruptions to manufacture caused by belt troubles and belt repairs.

The two following tables are taken from a report of Prof. C. H. Benjamin on "Tests of Leather Belting," Digest of Physical Tests, Vol. I, page 50.

TABLE VIII.

Series No.	Single or Double	Breaking Load lbs. per Sq. In.	Load per inch of width	Per cent of Elongation	Number of Samples
1	Single	4,770	905	9.88	13
1	Double	3,525	1,360	6.60	15
2	Single	5,107	1,008	7.20	15
2	Double	4,605	1,415	4.80	15
3	Single	3,730	841	10.18	14
3	Double	4,173	1,414	5.96	15
Average	Single	4,536	918	9.09	
Average	Double	4,101	1,396	5.79	

Average Strength 4300 pounds per square inch.

Single..... 918 pounds per inch of width.

Double.....1396 pounds per inch of width.

TABLE IX.

Length of Lap	Number of Rivets	Total Breaking Load	Load per Square inch Cross Sec.	Load per inch of width	Percent of Elongation
5.0	6	4,170	3,940	709.0	7.5
4.6	6	3,610	3,600	611.0	7.5
7.2	9	4,520	3,545	762.0	5.5
5.1	8	2,420	1,792	406.8	7.5
Cemented		4,380	5,560	1112.0	7.0

TABLE X.

Taken from Sellers' "Experiments on the Transmission of Power by Belting," Transactions of the A. S. M. E. Vol. VII.

Straight open belt $5\frac{1}{2}$ " wide by 7-32" thick and 34 ft. long, weighing 16 lbs., old but in good pliable condition, with hair side on C. I. pulleys 20 in. diam. running at 160 R. P. M. or about 800 ft. per min.

Sum of Tensions $T_2 - T_1$			$T_2 - T_1$ Working	T_2	T_1	$\frac{T_2}{T_1}$	Percentage of slip	Velocity of slip in F. P. M.	Arc of Contact	Coefficient of Friction		
Initial	Working	Final										
200	205	180	80	147.5	67.5	2.18	0.5	2.0	178	.261		
	210		100	155.0	55.0	2.82	0.9	3.6	177	.386		
	215		120	167.0	47.5	3.52	1.7	6.8	177	.407		
	220		140	180.0	40.0	4.50	3.0	12.0	176	.490		
	246		180	213.0	33.0	6.45	12.0	48.0	175	.610		
	300		120	210.0	90.0	2.33	.5	2.0	179	.270		
300	300		160	235.0	75.0	3.13	.8	3.2	179	.365		
	315		180	247.5	67.5	3.67	1.0	4.0	178	.418		
	320		200	260.0	60.0	4.33	1.7	6.8	178	.472		
	325		220	272.5	52.5	5.19	2.6	10.4	177	.545		
	340		240	290.0	50.0	5.80	3.8	15.2	177	.569		
	350		260	305.0	45.0	6.77	5.5	22.0	176	.623		
	360		280	320.0	40.0	8.00	8.6	34.4	176	.677		
	375		300	337.5	37.0	9.00	15.2	60.8	175	.719		
	400		420	385	200	310.0	110.0	2.82	.6	12.4	179	.386
			460		280	370.0	90.0	4.11	1.0	4.0	179	.452
			480		340	410.0	70.0	5.86	1.5	6.0	178	.509
510		400	455.0		55.0	8.27	2.2	8.8	177	.684		
535		440	487.5		47.5	10.20	4.5	18.0	177	.760		
560		480	520.0		40.0	13.00	8.4	33.6	176	.884		

Solid belt good for 4,300 pounds per square inch, or 918 pounds per inch of width.

Average riveted joint good for 622 pounds per inch of width.

Efficiency of joint—Riveted $622 \div 918 = 68$ per cent.

Efficiency of joint—Cemented $1112 \div 918 = 120$ per cent.

90. Thickness:—The thickness of belting is usually: single belt = $\frac{3}{16}$ inch; double belt = $\frac{5}{16}$ inch; and triple belt = $\frac{7}{16}$ inch.

91. Width by Common Method.—The width of a belt is always determined from the horse power formula. The simplest way of obtaining this is as follows:

Application:—Given a pulley running at 300 R. P. M. and transmitting 100 H. P. with a belt speed of 4000 F. P. M. What is the diameter of the pulley and the width of the belt?

$$V = 4000 = \pi D N; D = 4.24' = 51''$$

Take the ultimate strength of belting at 4000 #[" and the factor of safety at 10, then $f = 400$ #[". Let $T_2 = 3 T_1$, then $p = 400 - 133 = 267$ #[". This is equivalent to 50 pounds per inch in width of single belt and 84 pounds per inch in width of double belt and agrees very closely with Kent's recommendations. From the formula

$$\text{H. P.} = \frac{P V}{33000}; 100 = \frac{P \times 4000}{33000}; P = 825 \text{ pounds.}$$

$$\frac{P}{p} = \frac{825}{267} = 3.1[" = \begin{cases} 16.5'' \text{ single belt.} \\ 10'' \text{ double belt.} \end{cases}$$

92. Width by Nagle's Formula:—Another very satisfactory formula if it is desired to take into account the centrifugal tension and the arc of contact of the belt is that given in Kent, Page 878 by Nagel. (For proof of this formula see Par. 101.)

$$\text{H. P.} = C. v t w \left(\frac{f - .012 v^2}{550} \right) \quad (13)$$

$$C = 1 - 10^{-00758 a \phi}$$

a = degrees of belt contact.

Φ = coefficient of friction taken at .3 to .4

w = width of belt in inches.

t = thickness of belt in inches.

v = velocity in feet per second.

f = stress on belt per square inch of section.

Take f for leather belts = $\begin{cases} 400 \text{ for lapped and riveted belts.} \\ 275 \text{ for laced belts.} \end{cases}$

TABLE XI.

 Values of $C = 1 - 10^{-00758 a \phi}$ for different arcs of contact.

ϕ = Coeff. of Friction	DEGREES OF CONTACT = a										
	90	100	110	120	130	140	150	160	180	190	200
.15	.210	.230	.250	.270	.288	.307	.325	.342	.359	.376	.408
.20	.270	.295	.310	.342	.364	.386	.408	.428	.448	.467	.508
.25	.325	.354	.381	.5407	.432	.457	.490	.508	.524	.544	.582
.30	.376	.408	.438	.467	.494	.520	.544	.567	.590	.610	.649
.35	.423	.457	.489	.520	.548	.575	.600	.624	.646	.667	.705
.40	.467	.502	.536	.567	.597	.627	.649	.673	.695	.715	.758
.45	.507	.544	.579	.610	.640	.664	.692	.715	.737	.757	.802
.55	.578	.617	.652	.684	.713	.739	.763	.785	.805	.822	.858
.60	.610	.649	.684	.715	.744	.769	.793	.813	.832	.848	.877
1.00	.792	.825	.858	.877	.897	.913	.927	.937	.947	.956	.969

Applying this formula to the above problem we have for $a = 180^\circ$ and $\phi = .4$

$$100 = .715 \times 66.66 \times \frac{1}{18} w \left\{ \frac{400 - .012 (66.66)^2}{550} \right\}$$

$w = 10.5$ inches for double belt

or $w = 17.7$ say 18 inches for single belts.

93. Approximate Capacity of Belt:—A good short cut rule for finding the capacity of belting appeared in Power, November, 1903, page 621.

RULE—“To find the Horse Power which a given belt will transmit; Multiply the diameter of either pulley in feet by its revolutions per minute and by the width of the belt in inches. Multiply the above product by a value ranging from 3 for light single laced belts to 8 for heavy double cemented belts and point off three places from the final product.” If $C =$ constant to be chosen, then

$$\frac{C \times D \times R. P. M. \times w}{1000} = H. P. \quad (14)$$

Applying this formula to the above problem we have, if the belt is to be double thickness and is to have cemented joint

$$w = \frac{100 \times 1000}{4.24 \times 300 \times 8} = 10 \text{ inches.}$$

The chief difficulty in this will be to assume the proper constant. Experience will be necessary in determining this constant, but when it is fully established the rule becomes one of easy application.

All of the above belting rules for high speed belting, if applied to shop conditions must be modified decidedly. It must be remembered that most shop belts are laced or hooked, resulting in a much weak-

ened belt. The strength of a laced, or hooked joint may be taken from 30 to 50% of the value of a solid belt, consequently, if the same factor of safety be used, the value of f will be correspondingly smaller. In Nagles formula $f = 400$ for solid or cemented belts, and 275 for laced or hooked belts. No definite rules can be laid down governing shop belts, but the designer must take into account all the conditions, and make every belt a special case.

94. Standard Belt Widths:—The sizes of belts leading to machines and counters must conform to the pulley sizes as specified by the manufacturers. For the average machine shop with a variety of lathes, planers, milling machines, shapers, drill presses and the like, the belts range from 2 to 4 inches in width. With the introduction of the new self hardening tool steel, the cutting speeds, and the weight of metal removed per hour are being increased to such an extent as to require a revision of former sizes. This, of course, is due to the increased power required to run the machines.

The speed of belts leading to counters and machines varies from 400 to 1000 feet per minute; this is far below the most efficient velocity, but is made necessary because of the limitations on the pulley diameters.

Belts are manufactured according to the following widths: 1" to 3" by $\frac{1}{4}$ inch variations; 3" to 7" by $\frac{1}{2}$ inch variations; 7" to 24" by 1 inch variations; and 24" to 48" by 2 inch variations.

95. Perforated Belts:—The rapid approach of a belt to a pulley forms an air cushion which reduces the friction of the belt on the pulley. To avoid this, belts are sometimes perforated. Link belts have this advantage with the additional advantage of flexibility.

Perforated pulley rims are very common and serve the same purpose as perforated belts.

96. Belt Fastenings:—Cement, lacing leather, wire and hooks. For an endless belt, cement and rivets, cement and stitching, cement and shoe pegs, or cement alone are used.

It is always advisable to have a cemented joint on a high speed belt. Proper arrangement however, must be made for taking up the stretch.

Never throw a wide belt onto a pulley from the side, if it can be avoided, there is danger of unduly stretching that side. Cement the belt while in place over the pulleys.

The total stretch of new leather belting is from 6 to 8% of the original length.

97. Relation Between the Tension in a Belt and the Pressure on a Pulley:—In any belt at rest, Fig. 37, let T = tension in pounds per inch of width in each line, and p = normal pressure on the pulley rim per inch of width, then,

$$2 T = \int_{-\frac{\pi}{2}}^{\frac{\pi}{2}} p \, ds \cos \theta = \int_{-\frac{\pi}{2}}^{\frac{\pi}{2}} p R \cos \theta \, d\theta$$

$$2 T = p R \left[\sin \theta \right]_{-\frac{\pi}{2}}^{\frac{\pi}{2}} \quad \text{and} \quad p = \frac{T}{R} \quad (14)$$

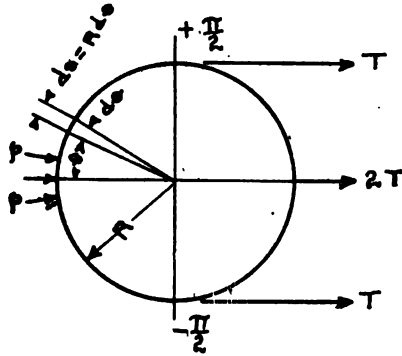


FIG. 37.

98. Relation Between the Tensions of the Tight and Loose Sides of a Working Belt in Terms of the Arc of Contact:—Let T_2 and T_1 be the tensions, respectively, of the tight and loose sides of the belt; also let ϕ be the coefficient of friction, B the arc of contact in π measure, and α the arc of contact in degrees, then from Fig. 38.

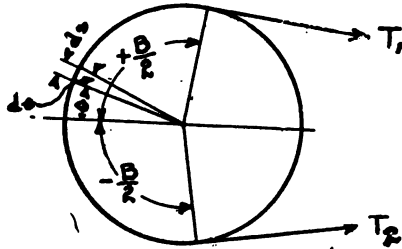


FIG. 38.

$$d T = d F = \phi p \, ds = \phi \frac{T}{R} R \, d\theta$$

$$\int_{T_1}^{T_2} \frac{d T}{T} = \int_{-\frac{B}{2}}^{\frac{B}{2}} \phi \, d\theta \quad \text{and} \quad \left[\log_e T \right]_{T_1}^{T_2} = \phi \left[\theta \right]_{-\frac{B}{2}}^{\frac{B}{2}}$$

$$\log_e \frac{T_2}{T_1} = \phi B \text{ but since } B = 3.1416 a \div 180$$

$$\text{we have } \log_e \frac{T_2}{T_1} = .01745 \phi a \text{ and } \log \frac{T_2}{T_1} = .00758 \phi a$$

$$\text{from which we get } \frac{T_2}{T_1} = 10^{-.00758 \phi a} \quad (15)$$

99. Horse Power Formula in Terms of the Maximum Pull on the Belt, T_2 and the Arc of Contact:—Having given the formulas:

$$(1), \text{ H. P.} = \frac{P V}{33000}; (2), P = T_2 - T_1; \text{ and } (3), \frac{T_2}{T_1} = 10^{-.00758 \phi a}$$

$$\text{obtain, } T_1 = \frac{T_2}{10^{.00758 \phi a}}; P = T_2 (1 - 10^{-.00758 \phi a}) = T_2 C$$

$$\text{and finally H. P.} = \frac{T_2 C V}{33000} \quad (16)$$

100. Horse Power in Terms of the Thrust on the Bearing, $2 T$ and the Arc of Contact:—In addition to the above three formulas (1), (2) and (3), assume $2 T_0 = T_2 + T_1$ (See Church Par. 171), and obtain

$$T_2 = T_1 (10^{.00758 \phi a}); 2 T_0 = T_1 (1 + 10^{.00758 \phi a});$$

$$T_1 = \frac{2 T_0}{1 + 10^{.00758 \phi a}}; P = 2 (T_0 - T_1) = 2 T_0 \left(1 - \frac{2}{1 + 10^{.00758 \phi a}}\right)$$

$$\text{and finally H. P.} = \frac{2 T_0 V}{33000} \left[\frac{10^{.00758 \phi a} - 1}{10^{.00758 \phi a} + 1} \right] \quad (17)$$

This formula may be especially useful in the design of machines for power measurements.

101. Horse Power Formula in Terms of T_2 , Arc of Contact, and Centrifugal Tension.—The centrifugal force of any piece of belting swinging around the centre of the pulley, causes it to push away from the pulley rim with a unit force p_o . This increases the value T_2 as previously given. From a former proof:

$$p_o = \frac{T_o}{R} = \frac{M v^2}{R}$$

where M = the mass of the leather and v = velocity in feet per second. Then if W = weight of a piece of leather 1 x 1 x 12 inches ($56 \div 144 = .388$ pounds), and V = velocity in feet per minute,

$$T_o = \frac{W V^2}{g 3600} = \left[\frac{.012 V^2}{3600} \right] \quad (18)$$

Again, if w = width of belt in inches, and t = thickness in inches, the pull on the tight side of the belt will be increased by the centrifugal tension and

$$f w t = T_2 + \frac{.012 V^2}{3600} w t. \text{ and}$$

$$T_2 = w t \left[f - \frac{.012 V^2}{3600} \right]$$

Then by substituting in (16) we have.

$$\text{H. P.} = \frac{C V w t}{33000} \left[f - \frac{.012 V^2}{3600} \right] \quad (19)$$

By comparison it will be seen that (19) is the same as (13) excepting F. P. M. is used instead of F. P. S.

102. Velocity at Which a Belt will Transmit its Maximum Power:—In (19), substitute $f = 400$, then if the first differential coefficient is taken equal to zero

$$\frac{d H P}{d V} = \frac{C w t}{33000} \left(400 - \frac{.036}{3600} V^2 \right) = 0.$$

$$V^2 = \frac{400 \times 36000}{.036} = 40,000,000.$$

$$V = 6300. \text{ Feet per minute.}$$

103. Velocity at Which the Tension, Due to the Centrifugal Force, will Equal the Working Strength:—

In (19) let $400 = .012 V^2 \div 3600$ then

$$V = 11000 \text{ feet per minute.}$$

104. Velocity at Which a Belt Will Break, Due to the Centrifugal Tension:—

In (19) let $4300 = .012 V^2 \div 3600$ $V = 35900$ feet per minute.

This value is about 50 per cent greater than the bursting speed of solid cast iron pulley rims.

105. References for Belt Transmission:—"Experiments on the Transmission of Power by Belting," Wm. Sellers & Co., Transactions of the A. S. M. E. Vol. VII; "Pulley and Belt Transmission," The Rockwood Mfg. Co., Indianapolis, Ind.; Nagle, "Horse-Power of Leather Belts," Transactions A. S. M. E., Vol. II; Taylor, "Notes on Belting," Transactions A. S. M. E., Vol. XV; Unwin, "Elements of Machine Design," Vol. I, pages 369-403.

106. Problems.—

(1). Find the horse power which may be transmitted by a double leather *shop* belt 6 inches wide, which passes over a pulley 5 feet in diameter, making 130 revolutions per minute.

The arc of contact between the belt and the pulley subtends an angle of 145 degrees; the coefficient of friction, ϕ , is known to be 0.38 and the tension on the tight side of the belt is not to exceed 35 pounds per inch of width.

(2). What is the maximum horse power which a 12 inch double belt 5-16 inch thick should be called upon to transmit as a *shop* belt. $a = 180$ degrees and $\phi = 0.40$?

(3). A certain *high speed* belt running 4000 feet per minute transmits 50 horse power between two shafts whose velocity ratio is 1. Find T_2 , T_1 and pull on bearings, with $\phi = 0.36$. Find T_2 , T_1 and pull on bearings, with $\phi = 0.54$. In this problem if a were changed to 270 degrees and $\phi = 0.36$, what would be the values of T_2 , T_1 and pull on bearings?

(4). A 6 inch single belt transmits 10 horse power from a pulley 36 inches in diameter, running at 200 revolutions per minute to a 24 inch pulley. Distance between shafts 8 feet, $\phi = 0.40$. If a spring balance is put in to pull the belt together when being laced, what pull will it show?

(5). A three-ply belt $\frac{5}{8}$ inch thick transmits 800 horse power from a fly wheel pulley 18 feet in diameter running at 80 revolutions per minute. $\phi = 0.40$, $a = 200$ degrees. With a working strength for the belt of 400 pounds per square inch, find the size of the belt taking into account centrifugal force. What per cent of the total stress is T_1 ? What per cent of the total stress is T_2 ? What per cent of the total stress is P ?

Rope Transmission.

107. Materials:—Power ropes are made from wires, manilla, hemp, cotton and rawhide. Wire ropes are used in cable work for hoists, conveyors and tramways, and fibrous ropes are preferred for pure power transmission. Of the kinds mentioned, *manilla* rope is the most common.

108. Velocity, Fibre Stress and Size of Ropes:—The *velocity* of a transmission rope should be about 5000 feet per minute for maximum efficiency. The *fibre stress* should be about 255 pounds per square inch. This is usually spoken of in terms of the diameter of the rope, as $200 d^2$. Rawhide may be taken $250 d^2$. The *size* of the rope may vary between $\frac{7}{8}$ inch for small powers and 2 inch for large powers. Probably the most common sizes are $1\frac{1}{8}$ to $1\frac{1}{2}$ inch.

The rope size may be governed, in some cases by the diameter of the smaller sheave, i. e., the diameter of the smaller sheave should be at least $36 d$.

109. Systems:—There are two systems of rope drives, Fig. 39. *A*, the *English* system, has a number of ropes running in parallel

over the same pulleys, each rope acting independently of the others. The *American* system, *B*, has but one rope looped continuously around the pulleys, as many times as there are desired strands in the drive. In the English system, the breaking of any one rope does

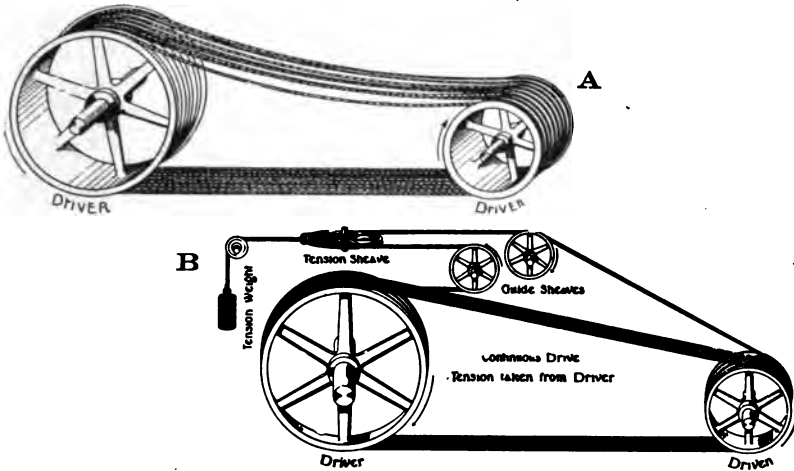


FIG. 39.

not seriously disable the system, while in the American, it would shut down the plant. On account of this, a close inspection of the ropes must be made at regular intervals to guard against accidents. The method of tightening the rope in the latter system, however, gives it an advantage over the former, in that there is less slippage between the ropes and the groove and consequently less wear on the ropes.

For details of design see "Rope Drives," by Flather, and "A Little Blue Book on Rope Transmission" by the American Manufacturing Company, New York City.

110. Simple Horse Power Formula:—In rope drives having slow speeds, and in such designs as do not require extreme accuracy, the following formula will be found satisfactory. Let $T_2 \div T_1 = 2$, (found to be the most satisfactory from experiments), V = velocity of rope in feet per minute, and N = number of ropes in the drive, then $P = T_2 \div 2 = 200 d^2 \div 2$ and

$$H. P. = \frac{100 d^2 N V}{33000} \quad (20)$$

111. Relation Between the Tensions of the Tight and Loose Sides of a Working Rope, in Terms of the Arc of Con-

tact.—Let θ be the angle of the groove, Fig. 40, (usually taken at 45°), then $2 \phi p_1 = \phi p (\operatorname{cosec} \frac{\theta}{2})$. Let $(\phi \operatorname{cosec} \frac{\theta}{2}) = \mu =$ equivalent coefficient of friction, then, substituting in (15) we have

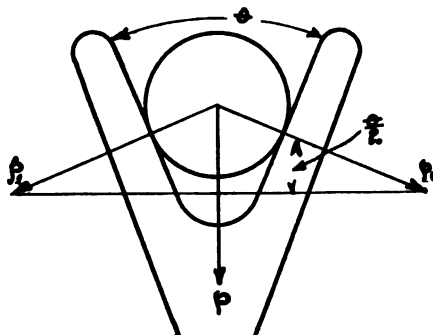


FIG. 40.

$$\frac{T_2}{T_1} = 10^{.00758 \mu a} \quad (21)$$

112. Horse Power Formula in Terms of the Maximum Pull on the Ropes, T_2 , and the Arc of Contact:—From (21), as in Par. 99, we have $T_2 (1 - 10^{-.00758 \mu a}) = T_2 K$ then

$$\text{H. P.} = \frac{T_2 K V}{33000} \quad (22)$$

113. Horse Power Formula in Terms of T_2 , Arc of Contact, and Centrifugal Tension:—As in belting we have, if the weight of one cubic inch = .034 pounds.

$$T = \frac{W}{g} \frac{V^2}{3600} = \frac{.32 d^2 V^2}{32.2 \times 3600} = \frac{.01 d^2 V^2}{3600} \quad (23)$$

If T_2 = tension in the ropes on the tight side, when not counting centrifugal tension, then the maximum fibre stress in the ropes under the new condition will be

$$f = T_2 + \frac{.01 d^2 V^2}{3600} \text{ or } T_2 = 200 d^2 - \frac{.01 d^2 V^2}{3600} \quad (24)$$

$$\text{then H. P.} = \frac{K d^2 V}{33000} \left(200 - \frac{.01 V^2}{3600} \right) \quad (25)$$

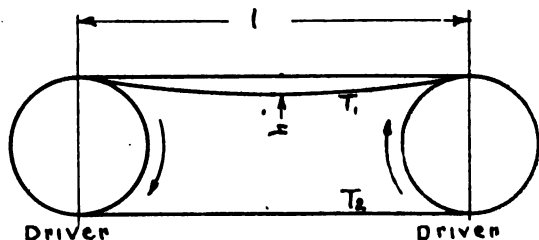


FIG. 41.

114. Relation Between the Tension and the Sag in a Rope or Belt:—The amount of sag in a flexible cord may be used in determining its tension. A rope is very nearly a flexible cord and will follow the same laws closely. The curve taken by a perfectly flexible cord hanging over pegs is a catenary, and is represented by

$$y = \frac{c}{2} \left[\left(e^{\frac{x}{c}} + e^{-\frac{x}{c}} \right) - c \right]$$

Unwin, Page 422. Where e is the base of the naperian system of logarithms. Referring to Fig. 31 this becomes

$$y + c = y^1 = \frac{c}{2} \left(e^{\frac{x}{c}} + e^{-\frac{x}{c}} \right) \quad (26)$$

If (26) be written in a decreasing series and collected to four terms

$$y' = \frac{c}{2} \left[2 + \frac{2x^2}{2c^2} \right] = c + \frac{x^2}{2c} \quad (27)$$

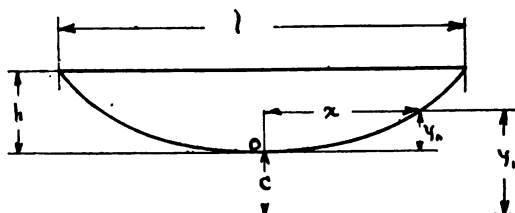


FIG. 42.

To determine c when $x = l \div 2$, $y' = c + l^2 \div 8c = c + h$ and $c = l^2 \div 8h$

It is the property of this curve that the tension at any point of the curve equals the weight of a portion of the rope having a length

y' , measured to the point selected. Expressed as an equation, $T = W y'$. From this we obtain

$$T = W (c + h) = W \left[\frac{l^2}{8h} + h \right] \quad (28)$$

where l = length of span in feet, and h = deflection of curve in feet. This equation is sometimes written

$$T = W (l^2 \div 8 h)$$

In the above, calculate T as T_1 of the slack side of the rope and then find T_2 from (21). This will give P in the general equation.

Example.—Power is transmitted by a rope-drive containing 20 fibrous ropes, each 1 inch in diameter. The velocity of the rope is 4500 feet per minute. The pulleys are 6 feet in diameter; they are at the same level and are 80 feet apart. The deflection of the slack side of the ropes is $4\frac{3}{4}$ feet. Find the horse-power transmitted and the deflection of the tight side of the ropes.

The tension in the slack side of one rope will be

$$T_1 = T = \frac{W l^2}{8 h} = \frac{.32 d^2 \times (80)^2}{8 \times 4.75} = 53.9\#$$

Assuming the coefficient of friction as, $\mu = 0.31$, with 180 degrees as the approximate angle of contact

$$T_2 = T_1 (10^{.00758 \times 0.31 \times 180}) = 2.647 T_1$$

Then $P = 1.647 T_1$ and $K = 0.6235$.

Applying equation (24) we find

$$f = 2.647 \times 53.9 + \frac{.01 \times 1 \times (4500)^2}{3600} = 199.45,$$

which shows the total stress of 199.45 pounds to be less than $200 d^2$, and therefore safe.

Substituting in formula (22).

$$\text{H. P.} = \frac{2.647 \times 53.9 \times 0.6233 \times 4500 \times 20}{33000} = 244.$$

Since each rope in the above calculation was allowed to take a maximum stress of 199.45 pounds on the tight side, we have a deflection on this side of

$$\frac{0.32 \times 1 \times (80)^2}{8 \times 199.45} = 1.28 \text{ feet.}$$

115. References for Rope Transmission:—Reuleaux, "The Constructor," pages 194-211; Low and Bevis, "Machine Drawing and Design," pages 154-168; Unwin, "Elements of Machine Design," Part I, pages 369-403; Jones, "Machine Design," Part II, pages 131-149; The American Manufacturing Company, New York, "A Little Blue Book on Rope Transmission;" Flather, "Rope Drives."

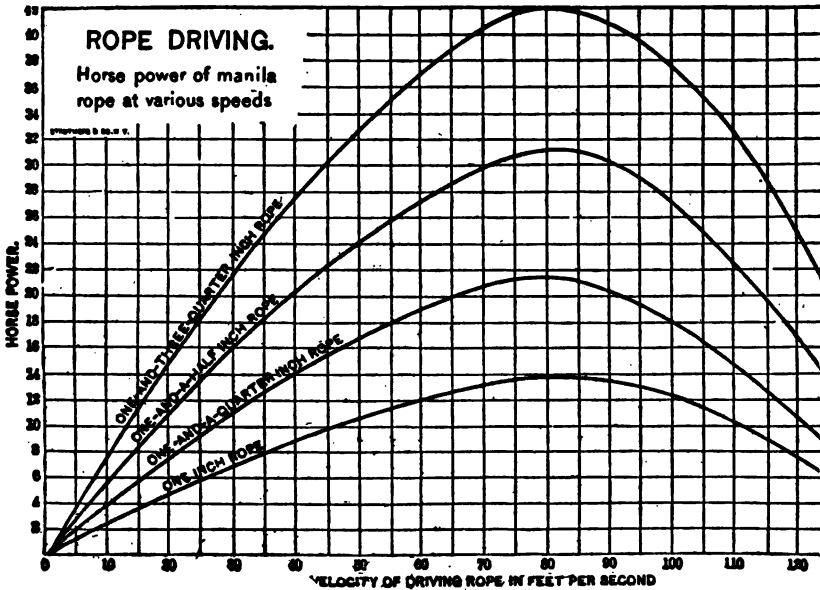


FIG. 43.

116. Problems:—

(1). Determine the maximum horse power that a $1\frac{1}{4}$ inch rope should be called upon to transmit. Assume $\phi = 0.28$, $a = 180$ degrees, and the angle of the groove, $\theta = 47$ degrees.

(2). (a). At what speed will a rope transmit its maximum horse power?

(b). At what speed will the stress due to the centrifugal force equal the allowable working strength?

(c). At what speed will the rope break due to the centrifugal force? Ultimate strength of manila rope = $7100 d^2$, ultimate strength of cotton rope = $4600 d^2$.

(3). Two shafts are 30 feet apart. One of them runs 90 revolutions per minute and carries a fly wheel pulley 14 feet in diameter from which it is desired to transmit 150 horse power to a pulley 42 inches in diameter on the other shaft. Assuming that the belt required is listed at \$4.80 per linear foot without discount of 50% and 10%; and that rope costs 15c per lb. with the additional expense of \$30 for tightener; estimate the relative cost of belt drive and rope drive to work under the above conditions.

Fly Wheels and Pulleys.

117. Fly Wheels and Fly Wheel Pulleys:—A fly wheel as a part of a machine serves the purpose of regulating the speed of the machine by alternately storing up and giving out work as the speed of rotation is increased or diminished. In addition a fly wheel may also serve the purpose of a belt pulley.

In designing the ordinary fly wheel pulley, i. e. a wheel which is to serve the double purpose of fly wheel and pulley, the stresses produced in the parts of the wheel by centrifugal force are usually neglected, however, the following things must be known: (1), the velocity of the rim; this is given in feet per minute of the belt velocity or in diameter and the revolutions per minute of the pulley; (2), the horse power transmitted; (3), the weight of the pulley rim; (4), single or double belt. Having given the above data with the knowledge of the use to which the pulley will be put, the following would be the method of procedure.

Given a fly wheel pulley with $D = 50''$; revolutions per minute $= 250$; horse power $= 75$; weight of rim $= 650\#$. Find the width of the belt and of the pulley rim and all sizes relating to the rim, arms, hub, shaft, and fastening.

118. Rim:—Allowing an arc of contact of 180° and a coefficient of friction of .4 we have, Par. 92, $w = 9.4$, say 10 inches. Taking the pulley rim one-half inch wider than the belt gives the width of the pulley face $= 10.5$ inches.

Let the thickness of the rim be t inches, then the weight of the rim becomes

$$W = \pi (D - t) .26 t w$$

$$650 = 3.1416 (50 - t) \times 10.5 \times .26$$

$$t = 1.56''$$

This assumes that the radius of gyration extends to a point half way between the inner and outer surfaces of the rim. This assumption is not theoretically correct, but for all practical purposes in the design of fly wheels, with comparatively thin rims, the error is so slight as to be negligible. Where a fly wheel is expected only to regulate speed and no belt is to be used in connection with it, the rim is made more nearly square in section, and, in such cases, the radius of gyration, R , should be taken at its true value,

$$\sqrt{\frac{R_1^2 + R_2^2}{2}}$$

where R_1 and R_2 are the external and internal radii of the cylinder. The above calculation also assumes that all the weight of the wheel

is centered in the rim, and that the arms and hub need not be considered. This is true for most fly wheels, for whatever purpose, since the rim, being the part farthest removed from the axis of rotation, will have the highest velocity and consequently possesses a greater amount of accumulated work than other parts of the same weight. Then too, in common practice the rim is heavy as compared with other parts of the wheel. See Church, "Mechanics of Engineering," Par. 106.

Where a fly wheel is to be designed to meet the needs of a "balance wheel" it becomes necessary to determine the weight, W , of the rim, in pounds. To do this the following items must be considered:—

R = radius of gyration of wheel in feet.

N = mean speed of wheel in revolutions per minute.

N_1 = maximum speed of wheel in revolutions per minute.

N_2 = minimum speed of wheel in revolutions per minute.

$n = \frac{N}{N_1 - N_2}$ = co-efficient of fluctuation of speed,

H' = work to be stored up by wheel, in changing its speed from the minimum to the maximum,

H = total work done in one revolution,

$r = \frac{H'}{H}$

$$H' = \frac{2 W \pi^2 R^2 (N_1^2 - N_2^2)}{g \times (60)^2}$$

Therefore,
$$W = \frac{900 g n r H}{\pi R^2 N^2}$$

The value of r will of course depend entirely upon the individual problem, while values of n vary widely in machines for different purposes, being as low as 20 to 30 for punching and shearing machines, and running up to 150 for direct connected high speed steam engines and dynamos.

With the weight of rim determined it becomes an easy matter to obtain its cross section from the known weight per cubic inch of the material used, and the desired proportion of width to thickness. Or width of the rim may be assumed and the thickness calculated.

119. Arms:—Select the shape of the arm section, Fig. 44, and calculate the arm as a beam under flexure. If this section is oval as shown in Fig. 44, the formula becomes

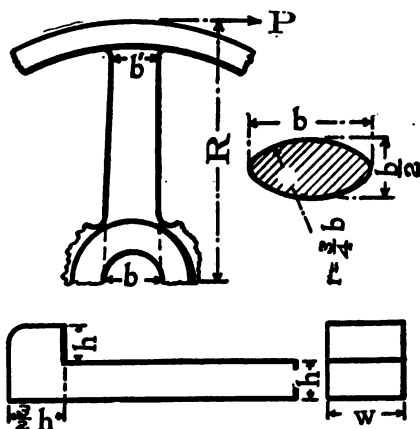


FIG. 44.

$$\frac{P R}{N} = .05 b^3 f.$$

where P = tractive force due to belt, or equivalent tractive force due to stored-up energy given out by fly wheel;

N = number of arms carrying load. Tests conducted under actual working conditions of belt indicate that all the load is taken by one-half the pulley arms while in a fly wheel without belt all arms will be equally loaded.

f = allowable fibre stress of the metal, 2550 pounds per square inch where the load is carried by one-half the arms, or 1500 pounds per square inch where load is taken equally by all the arms.

The values for the fibre stress are taken low to make allowance for the unknown stresses due to centrifugal force.

The dimensions of an arm at the rim should not be taken less than two-thirds of the corresponding dimensions at the hub. In the above application for a six arm pulley, considering one-half the arms to take the entire load, we have;

$$b = \sqrt[3]{\frac{P R.}{3 \times .05 \times 2500}} = 3.7, \text{ say } 3\frac{3}{4}''$$

From this value of b the other values may be obtained.

In designing pulley arms the size b at the center is sometimes taken equal to the diameter of the shaft and the other proportions as given above. This does not sufficiently account for all the straining actions of the belt and cannot be recommended further than as a check on the calculations.

Straight arms are preferred to curved arms and they should have good, well rounded fillets next to the hub and rim.

The section of the arm near the hub and the section near the rim are always similar. Of the sections shown in H, Fig. 48, 1 to 6, 2 is the one found in general use on medium sized wheels; large wheels having arms shaped like 4, 5 and 6. Sections 1 and 2 have an advantage over the others in wind resistance. This in high speed wheels is worthy of consideration.

120. Shaft:—If the pulley or fly wheel is to be used on an engine shaft, the diameter of the hole will be

$$d = 7.3 \sqrt[3]{\frac{\text{H. P.}}{\text{R P.M.}}} = 4.88, \text{ say } 4\frac{7}{8} \text{ inches. See Par. 69.}$$

If it is to be used on a machine shaft the diameter of the shaft, or the hub bore, would be determined for the condition of combined twisting and bending, see Par. 68.

121. Hub:—(See also Par. 126). It is common practice to take the *diameter* of the hub as twice the diameter of the shaft, this would be $9\frac{3}{4}$ inches. The *length* of the hub should bear some relation to the width of the rim or to the shaft diameter. A ratio sometimes used is twice the shaft diameter. A more satisfactory figure would be from two thirds the width of the rim to the width of the rim.

122. High Speed Fly Wheels and Pulleys.—In any wheel moving at a high rotative speed there is a uniform expansion of the rim which tends to increase its diameter, thus producing a uniform tensional force on the rim section. There is also an increased length of the arm, but the expansion of the arms not being sufficient to equal the increased rim diameter, there is a bending moment produced in the rim between the arms. The uniform tensional stress and the bending stress may each be determined for any section of the rim and then combined as follows into a fibre stress f .

123. Stresses in Fly Wheel Rims Produced by Centrifugal Force:—The magnitude of the centrifugal force is given by the equation

$$\text{C. F.} = \frac{W v^2}{g R} \quad (29)$$

where W = weight in pounds, v = velocity in feet per second, R = radius in feet, $g = 32.2$. Let t = thickness of rim in inches, w = width of rim in inches, l = length of unit arc in inches; then if one cubic inch of cast iron weighs .26 pound, we have

$$W = 0.26 w t l$$

For a unit section parallel with the rim, 1 inch \times 1 inch, this becomes

$$W = .26 t$$

substituting this value of W in (29) gives C. F. in pounds per square inch as

$$\text{C. F.} = p = \frac{0.26 t v^2}{32.2 R} \quad (30)$$

For a cylinder subjected to an internal pressure of p pounds per square inch we have $f = p D \div 2 t$, where f = tension produced in the rim in pounds per square inch, D = diameter in inches and t =

thickness in inches. Substituting for p the value given by (30) we have

$$f = \frac{0.26 t v^2 \times 2 \times 12 \times R}{32.2 R \times 2 \times t} = \frac{v^2}{10.3} \quad (31)$$

from which we see that if we neglect the effect of the arms, the stress produced by the centrifugal force in a pulley rim varies as the square of the velocity and does not depend on the dimensions of the pulley. This stress in the rim will be called the centrifugal tension and is the stress that is commonly taken into account by designers. The stresses *actually existing* in the rim when in motion are very difficult to determine. It is safe to assume that they are as follows:

(1) A direct tensile stress which is a portion only of the centrifugal tension.

(2) Stresses due to the bending of the portion of the rim between two adjacent arms.

To explain more fully, suppose that *befg* Fig. 45 *A*, is the center of the rim of a pulley when at rest. If the pulley is put in motion and we neglect the effect of the arms, centrifugal tension will in-

Showing Effect of Centrifugal Force on
Fly-Wheel Rim.

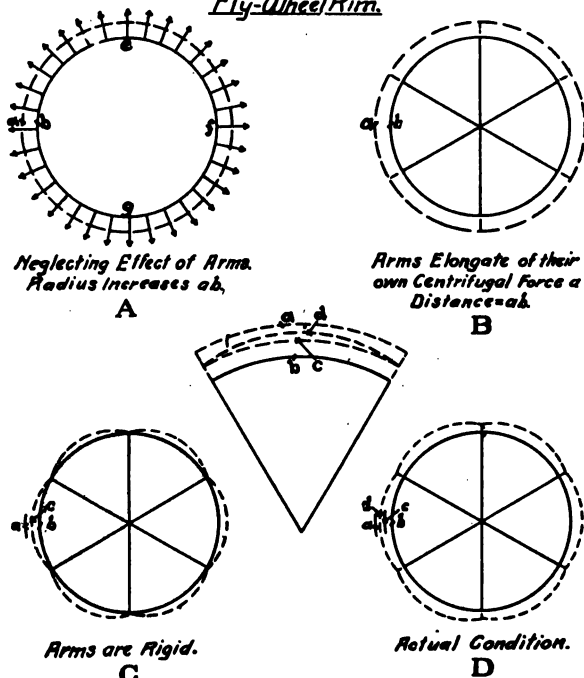


FIG. 45.

crease the circumference as shown by the dotted line; but if we take into account the arms and regard them as allowing the full expansion of the rim, it is evident that they must lengthen an amount ab as shown at B. On the other hand if the arms were inextensible we would have the condition of a beam uniformly loaded as shown at C. The actual shape of the rim, however, will be somewhere between B and C as shown in D, where the arms elongate a distance bc and the rim bends an amount cd . Hence the statement for the stresses which actually exist in the rim of a pulley as given above when put in a formula is:

$$f = \frac{T}{A} \pm \frac{M}{Z} \begin{cases} - = \text{outside of rim} \\ + = \text{inside of rim.} \end{cases} \quad (32)$$

where f = stress in pounds per square inch at any point.

T = tension in rim due to centrifugal force, in pounds.

A = area of rim section in square inches.

M = bending moment.

Z = Modulus of section.

To determine T and M it is necessary to first find the pull F exerted on each arm such that $F \div 2$ is the shear at the point of support. It is very evident that

$$F = C \frac{W}{g} v^2 \quad (33)$$

where C is a constant to be determined. Now it can be shown analytically* that when W = weight of a cubic foot of iron = 450 pounds and $g = 32.2$ the value of C is

$$C = \frac{1}{3} \left\{ \frac{2 - \left(\frac{r_1 - r_2}{R} \right)^2 \left(\frac{r_1 + \frac{1}{2} r_2}{R} \right)}{\frac{1}{A'_1} \left(\frac{r_1 - r_2}{R} \right) + \frac{1}{2 A' a}} \right\} \quad (34)$$

r_1 = distance in feet from center of hub to outer end of arm.

r_2 = radius of hub in feet.

R = distance from center of rim to center of hub in feet.

A' = area cross section of rim in square feet.

A'_1 = area cross section of arms in square feet.

a = $\frac{1}{2}$ angle between arms in π measure.

Φ = angle between arm and a variable point.

Using the value of F thus determined it gives

*For complete analysis see Vol. XVI Proceedings A. S. M. E. "Stresses in Rims and Rim-joints of Pulleys and Fly Wheels." Lanza.

$$T = \frac{W}{g} A' v^2 - \frac{F \cos (a - \Phi)}{2 \sin a} \quad (35)$$

$$M = \frac{F R}{2} \left(\frac{1}{a} - \frac{\cos (a - \Phi)}{\sin a} \right) \quad (36)$$

and f the maximum stress in the rim becomes known by substituting in equation (32). Applying the above to a wheel as shown in Fig. 46 we have from formula (34), if $v = 60$ F. P. S.

$$C = .0189$$

Substituting this value in (33) we have

$$F = 951.$$

Substituting this value in (35) for each of the angles 0, 10, 20 and 30 degrees we have

	0°	10°	20°	30°
$T =$	4766	4696	4653	4639
and $\frac{T}{A} =$	298	293	291	289

Substitute the value F as given above, also in (36) and obtain

	0°	10°	20°	30°
$M =$	146.29	24.63	-50.43	-75.7
$\frac{M}{Z} =$	331	55.	-113.	-170

If now, we substitute the values $\frac{T}{A}$ and $\frac{M}{Z}$ in equation (32) it gives

	0°	10°	20°	30°	
$f =$	629	348	178	119	Inside fibres.
$f =$	-33	237	404	461	Outside fibre.

Plotting the values $\frac{T}{A}$, $\frac{M}{Z}$ and f for the inner and outer fibres of the rim we see, Fig. 47, that at about thirteen degrees from each arm the stresses in the outer and inner fibres are equal. It is evident then that the rim joints of sectioned rims might theoretically, be located at these points.

However, moving the large metal mass of the joint changes the bending moment, and destroys the original, assumed conditions. From Prof. Benjamin's experiments it has been shown that the best place for the joint is at the end of an arm, and modern belt flywheels of the most careful designers show the joints so placed.

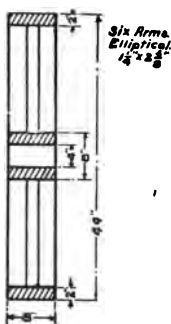


FIG. 46.

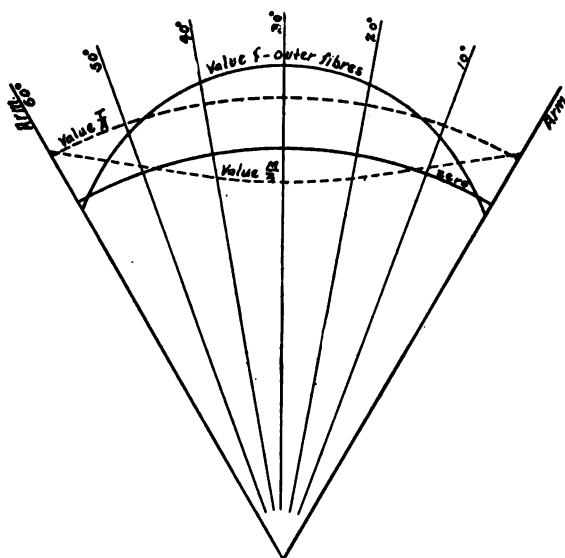
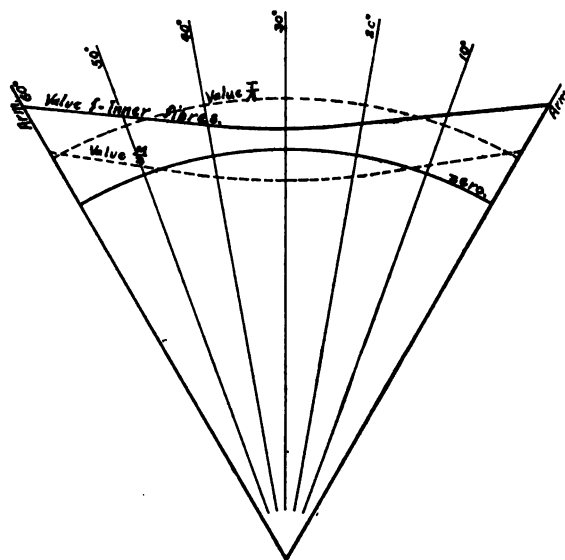


FIG. 47.

124. Forms of Belt Pulley Rims:—Belt pulley rims and fly wheel pulley rims are usually made of one of the forms *A*, *B*, and *C*, Fig. 48. Under favorable conditions with a thick rim section *A*

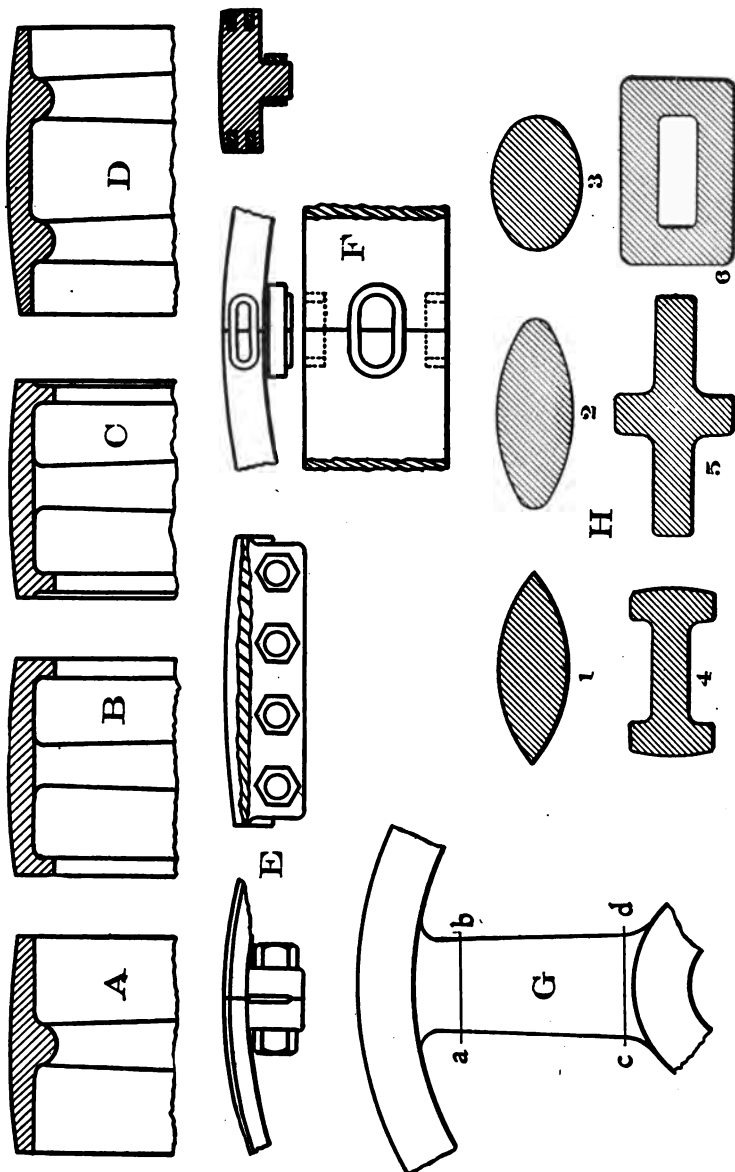


FIG. 48

would be satisfactory; but if the rim is wide and not very thick the cross strain on the rim due to its centrifugal force may endanger it and flanges as at *B* and *C* are added to support the outer edge. Where the rim is of unusual width, two sets of arms may be provided as at *D*.

Solid rims have unknown stresses in the metal due to the cooling action in the mold. These may be somewhat relieved by making the rim in sections and bolting the sections together. Other advantages are to be gained by sectioning the pulleys; the chief ones being, that of handling in its production and shipping, and the ease of adjustment to the shaft without removing the shaft from its bearings.

Two common methods of fastening the sections together are shown in *E* and *F*. *E* is the simplest and is more often found than *F* but it is not so strong. In the Trans. A. S. M. E. Vol. 20, will be found a report of a series of experiments by Prof. Benjamin on the explosion of pulleys running at high speeds in which he brings out the following conclusions:

(1) Fly wheels with solid rims, of the proportions usual among engine builders and having the usual number of arms, have a sufficient factor of safety at a rim speed of 100 feet per second if the iron is of good quality and there are no serious cooling strains. In such wheels the bending due to the centrifugal force is slight and may be disregarded.

(2) Rim joints midway between the arms are a serious defect and reduce the factor of safety very materially. Such joints are as serious mistakes in design as would be a joint in the middle of a girder under a heavy load.

(3) Joints made in the ordinary manner, with internal flanges and bolts, are probably the worst that could be devised for this purpose. Under the most favorable circumstances they have only about one fourth the strength of a solid rim and are particularly weak against bending. In several joints of this character on large fly wheels, calculation has shown a strength less than one fifth that of the rim.

(4) The type of joint known as the link or prisoner joint is probably the best that could be devised for narrow rimmed wheels not intended to carry belts, and possesses when properly designed, a strength about two thirds that of the solid rim.

125. Forms of Rope Pulley Rims:—Rope pulley rims take the forms shown by Fig. 49. *A* and *B* are typical sections of transmission pulley rims for fibrous ropes; *C*, is a section of a rim for wire rope transmission; and *D*, is the section of an idler pulley rim. Attention is called to the fact that the fibrous ropes *wedge* in the grooves and wire ropes rest upon the bottom of the groove. Also, that the wire rope groove has a hard wood bottom.

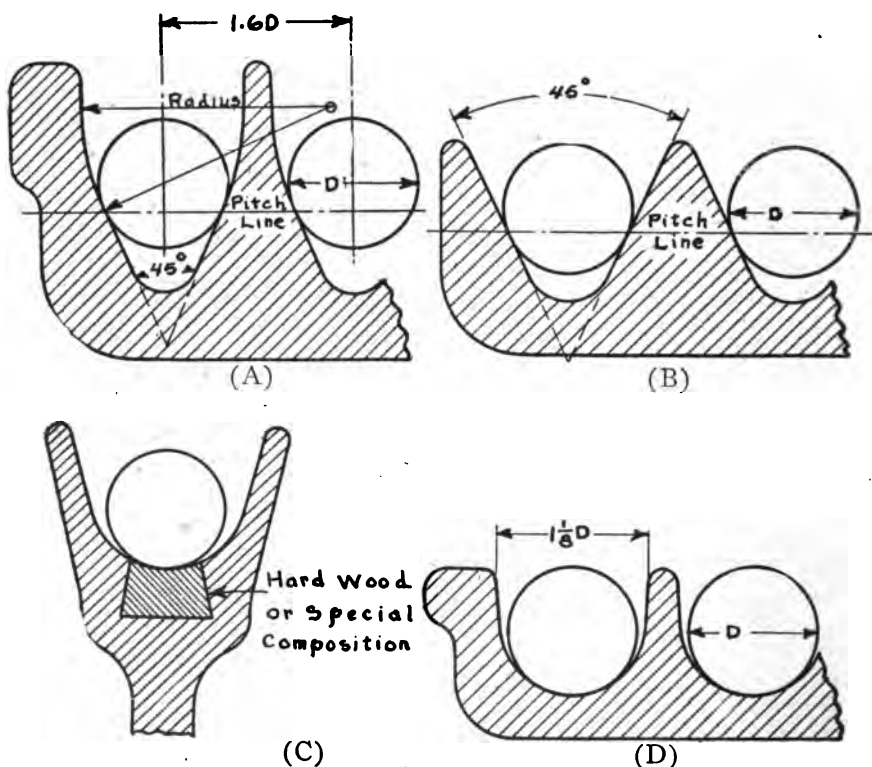


FIG. 49.

126. Hub Forms:—Hubs in like manner with the rims are made either solid or sectioned. Many wheels have solid rims and split hubs, thus with the flexibility at the hub reducing the cooling strains in the wheel and providing an easy adjustment to the shaft. In Fig. 50 *I*, shows a typical solid hub section, *J*, shows a single cut hub and *K* a double cut hub.

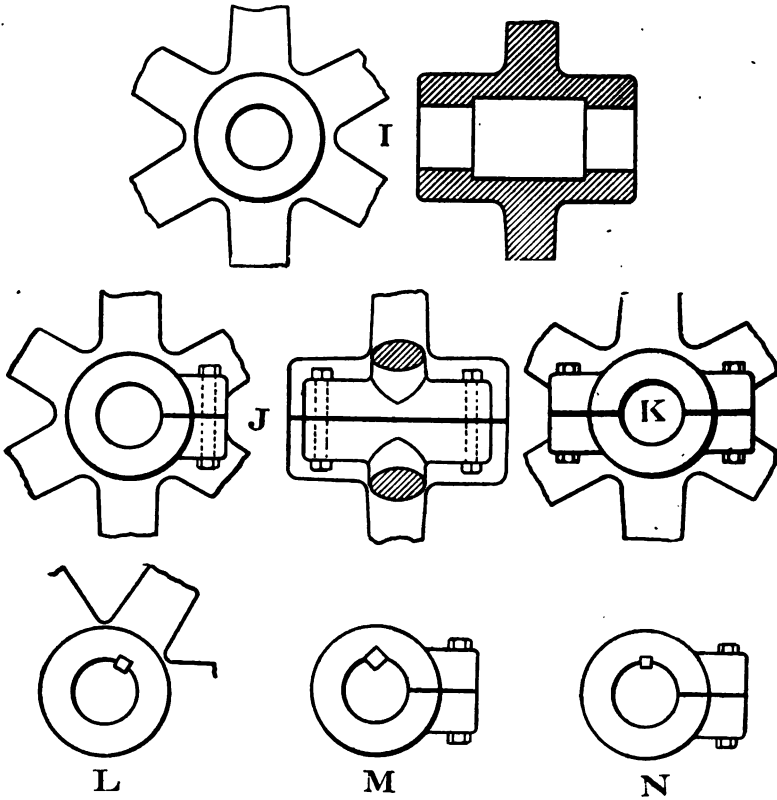


FIG. 50.

A sectioned hub makes it easy to remove the key and wheel from the shaft. In fitting keys to pulleys they are generally placed under an arm. In a solid hub any arm will do, but in split hubs a location should be taken approximately at right angles to the cut where the tension on the bolt produces a clamping action on the key between the hub and the shaft. Keys are sometimes set diagonally; this however, is not to be recommended.

127. Shafting Pulleys:—Pulleys for use on line and counter shafts are made of wood, paper, cast iron and steel. Those used on machines are generally made of cast iron.

The wood pulley is generally split in two parts and bolted around the shaft, being held to the shaft by friction. Wood pulleys have found a large use in factories because of the ease with which they can be applied to or removed from the shaft. They can be had of any standard diameter and width of face. Any one pulley can be fitted to shafts of varying diameters by the use of bushings.

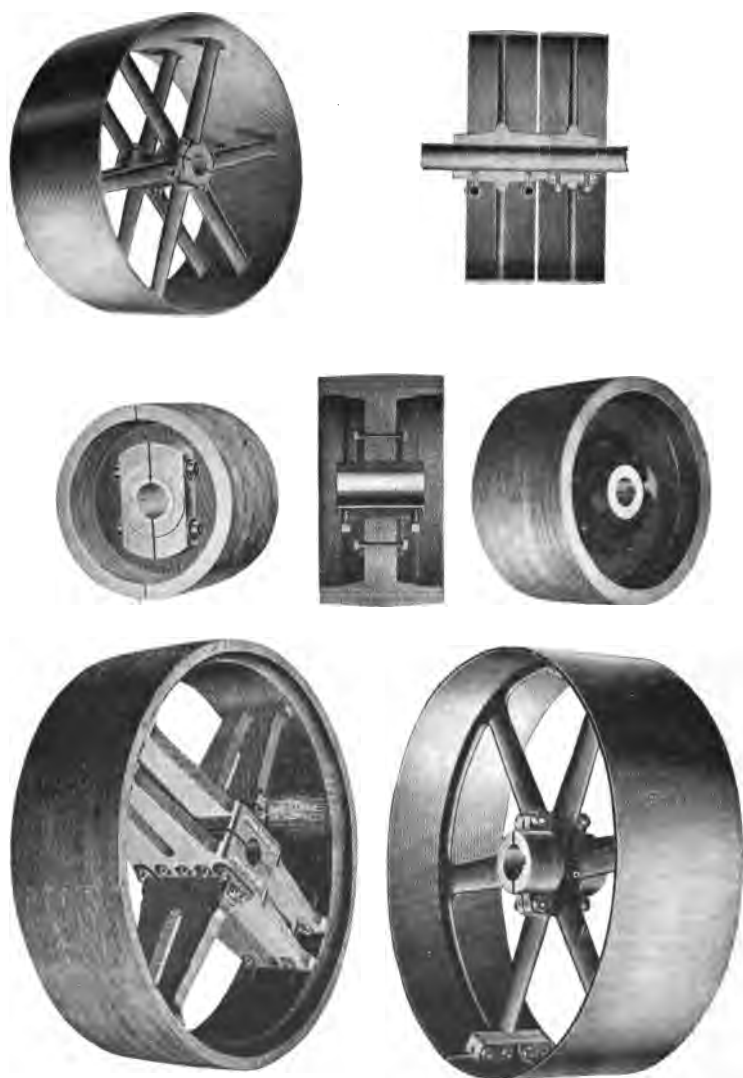


FIG. 51.

The cast iron pulley is the standard pulley. It is usually made solid although a split pulley can be obtained by special order. The following notes will apply to small, solid cast iron pulleys.

The diameter D usually varies by inches.

The width of face $B = \text{width of belt} + (\frac{1}{4}" \text{ to } \frac{1}{2}")$

The thickness of the rim, $t = \frac{D}{200} + \frac{1}{4}"$

Pulleys should be coned $\frac{1}{8}"$ to $\frac{1}{16}"$ in diameter for each inch of width, the narrower faces having the heavier cone.

Tight and loose pulleys used for shifting belts do not need coning.

Taper the inside of the rim and have good fillets between rim and arms.

The section of the arm can be calculated as shown in Par. 119. As a matter of fact, however, the arms of small pulleys are made much heavier than would be calculated from the forces involved. The chief reason for this is that pulleys may be used for any kind of service and must be designed to withstand the heaviest loads.

Concerning the number of arms in a pulley it may be said that in general, pulleys below 12 inches in diameter have 4 arms, and above 12 inches, 6 arms.

128. References for Fly Wheels and Pulleys:—Benjamin, "Machine Design," pages 166-183; Jones, "Machine Design," Part II, pages 243-272.

Hubs, Keys and Couplings.

129. Standard Hubs:—The following hub sizes of various machine parts, representing the current practice of four manufacturing companies, was reported in the American Machinist, January 14, 1904.

Diameter of Hubs where $d = \text{Diameter of Shaft.}$

	Cast Iron	Cast Steel.
Heavy, very great shock	2 d	$1\frac{3}{4} d + \frac{1}{8}"$
Standard medium shock	$1\frac{3}{4} d + \frac{1}{8}"$	$1\frac{5}{8} d + \frac{1}{16}"$
Light, no shock	$1\frac{5}{8} d + \frac{1}{8}"$	$1\frac{1}{2} d + \frac{1}{4}"$

Length of Hubs.

Truck wheels	2 d to $2\frac{1}{4} d$	Gear wheels	$1\frac{3}{4} d$ to $2\frac{1}{4} d$
Hand wheels	$1\frac{1}{2} d$ to 2 d	Bearings	3 d to 4 d
Levers	$1\frac{1}{2} d$	Pulleys	Face

130. Keys:—A key for a large pulley should be made according to the shape shown in Fig. 44.

If d = diameter of shaft
 then $W = \frac{1}{5} d$ for a six inch shaft.
 $W = \frac{1}{4} d$ for a two inch shaft.
 $h = \frac{1}{4} W$ to $\frac{3}{4} W$ for a six inch shaft.
 $h = \frac{3}{4} W$ to W for a two inch shaft.

The key should *taper* $\frac{1}{8}$ inch to $\frac{3}{16}$ inch per foot of length.

The following tables XII, concerning the size of hub keys and key seats, are in current use by reputable manufacturing firms.

TABLES XII.

DIAMETER OF SHAFT.	SIZE OF KEY SEAT.	
	WIDTH "A."	DEPTH "B."
Inches.	Inches.	Inches.
$\frac{3}{4}$ to $1\frac{1}{4}$ inclusive.....	$\frac{1}{4}$	$\frac{1}{8}$
$1\frac{5}{8}$ to $1\frac{3}{4}$ ".....	$\frac{3}{8}$	$\frac{3}{16}$
$1\frac{7}{8}$ to $2\frac{1}{4}$ ".....	$\frac{1}{2}$	$\frac{1}{4}$
$2\frac{3}{8}$ to $2\frac{3}{4}$ ".....	$\frac{5}{8}$	$\frac{1}{8}$
$2\frac{7}{8}$ to $3\frac{1}{4}$ ".....	$\frac{3}{4}$	$\frac{3}{8}$
$3\frac{1}{8}$ to $3\frac{3}{4}$ ".....	$\frac{7}{8}$	$\frac{7}{16}$
$3\frac{5}{8}$ to $4\frac{1}{4}$ ".....	1	$\frac{1}{2}$
$4\frac{5}{8}$ to $4\frac{3}{4}$ ".....	$1\frac{1}{8}$	$\frac{9}{16}$
$4\frac{7}{8}$ to $5\frac{1}{4}$ ".....	$1\frac{1}{4}$	$\frac{5}{8}$
$5\frac{1}{8}$ to $5\frac{3}{4}$ ".....	$1\frac{3}{8}$	$\frac{11}{16}$
$5\frac{5}{8}$ to $6\frac{1}{4}$ ".....	$1\frac{1}{2}$	$\frac{3}{4}$

STEEL KEYS AND SHAFTS.

Size of Shaft.....	$1\frac{1}{8}$ - $\frac{3}{4}$	$1\frac{3}{8}$ - $\frac{7}{8}$	$1\frac{5}{8}$ -1	$1\frac{7}{8}$ - $1\frac{1}{8}$	$1\frac{9}{8}$ - $1\frac{1}{2}$	$1\frac{9}{8}$ - $1\frac{7}{8}$
Size of Key.....	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{7}{32}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$
Size of Shaft.....	$1\frac{1}{8}$ - $2\frac{3}{8}$	$2\frac{7}{8}$ - $2\frac{7}{8}$	$2\frac{1}{8}$ - $3\frac{1}{2}$	$3\frac{9}{8}$ -4	$4\frac{1}{8}$ - $4\frac{5}{8}$	$4\frac{1}{8}$ - $5\frac{1}{4}$
Size of Key.....	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$

131. Couplings:—Pieces of shafting are held together by couplings. The figures in tables XIII show the common forms. The solid sleeve coupling is very little used except on very light work, the compression coupling and the double cone vise coupling are used on shafting lines, the flange coupling is used on heavy transmission lines such as principal shafts and jack shafts, and the jaw clutch coupling is used on shafts requiring frequent disconnections. Tables XIII, from catalog data, will be of value in proportioning the various parts.

TABLES XIII.

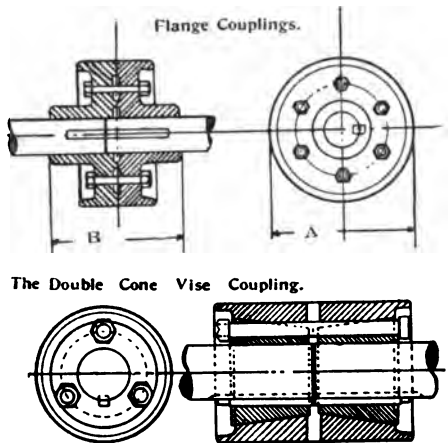


FIG. 52.

FLANGE COUPLING

Size of Shaft in Inches	A	B
1 1/8	6	4 1/2
1 1/4	6 5/8	5
1 1/2	7 1/8	5 5/8
1 3/4	7 3/4	6 1/8
2	8 3/8	6 5/8
2 1/4	9	7 1/4
2 1/2	9 1/2	7 5/8
2 3/4	10 1/8	8 1/4
3	10 1/2	8 5/8
3 1/4	11 1/8	9 1/8
3 1/2	11 3/8	9 5/8
3 3/4	12 1/8	10 1/4
4	13	11
4 1/4	13 3/8	11 5/8
4 1/2	14 1/8	12 1/4
4 3/4	14 3/8	12 5/8
5	15 1/8	13 1/4
5 1/4	17	14 1/4

THE DOUBLE CONE VISE COUPLING

Size of Shaft	Approx Diameter of Coupling	Approx Length of Coupling	Width of Key-Seat	Depth of Key-Seat	Length of Keys	Approx Weight Each in Pounds
Inches	Inches	Inches	Inches	Inches	Inches	
1 1/8	3 3/4	5 1/4	1/8	3/16	2 1/8	13
1 1/4	4 1/8	6 1/8	1/8	3/16	2 1/8	20
1 1/2	4 5/8	6 5/8	1/8	3/16	2 3/8	33
1 3/4	5 1/4	7 1/8	1/8	3/16	3 1/8	42
2	5 5/8	8 1/4	1/8	3/16	3 3/8	60
2 1/4	6 1/8	9 1/8	1/8	3/16	4	72
2 1/2	7	10	1/8	3/16	4 1/8	100
2 3/4	7 3/8	11	1/8	3/16	4 3/8	120
3	8 1/4	12	1/8	3/16	4 3/8	150
3 1/4	9 1/8	13	1/8	3/16	5 1/8	215
3 1/2	10	14 1/4	1/8	3/16	5 3/8	310
3 3/4	11 1/8	16	1/8	3/16	6 1/8	400
4	12 1/8	18	1/8	3/16	7 1/8	570
4 1/4	13 1/8	19 1/2	1/8	3/16	8 1/8	700
4 1/2	14 1/8	21	1/8	3/16	8 3/8	

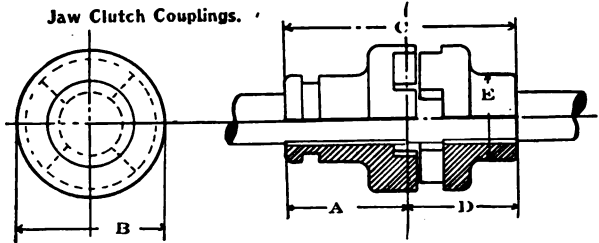


FIG. 53.

JAW CLUTCH COUPLING

Size Inches	A Inches	B Inches	C Inches	D Inches	E Inches
1½	2	3	5½	2	3¾
1¾	2½	4½	8	3½	4¾
1⅞	3¾	6	10½	4	6
2⅞	4½	7½	12¾	5	7½
2⅞	5½	9	14¾	6	8½
3⅞	6½	10½	17¾	7	10
3⅞	7	12	19¾	8	11
4⅞	8	13½	22	9	12½
4⅞	8½	15	24	10	13½
5¾	10½	18	28½	12	16
5¾	12	21	28½	12	16½

Solid Sleeve Couplings.
For Light Shafting.

Ribbed Compression Couplings.



FIG. 54.

132. Counter Shafts:—Counter shaft journals and boxes differ materially from journals and boxes on line shafts. Counter shafts are sometimes made from pieces of cold rolled shafting without being turned at the journals, the latter being located by set col-

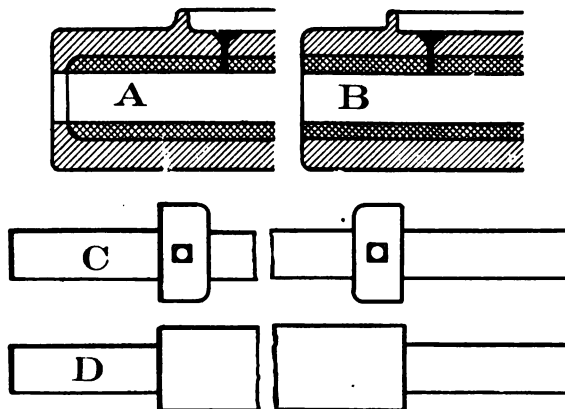


FIG. 55.

lars as at C, Fig. 55. The ordinary counter-shaft however, is turned smaller at the journals thus forming a shoulder and avoiding the necessity of a set collar, as at D.

Journal boxes for counter shafts are usually solid and babbitted, the babbitt being from ⅛ inch to ¼ inch thick. If the box is to be used with a set collar the babbitt may come flush with the end of the casting as at B, but if the journal is shouldered the metal of the box should extend down *nearly* to the shaft as at A. In any case

the end thrust of the shaft must be taken up *iron to iron*. Every counter shaft should have a small end play to avoid cutting the end of the box.

The following may be of value in finding the size of a counter shaft where it is to bear a certain relation to the width of the belt.

2"	Belt.....	$1\frac{3}{8}"$	shaft.
$2\frac{1}{2}"$	"	$1\frac{5}{8}"$	"
3"	"	$1\frac{7}{8}"$	"
$3\frac{1}{2}"$	"	$1\frac{9}{8}"$	"
4"	"	$1\frac{1}{8}"$	"
5"	"	$1\frac{1}{8}"$	"

133. Set Collars.—In all lines of shafting it is necessary to provide a means of taking up the end thrust of the shaft. This is done by set collars next to the Journal boxes. *A*, Fig. 56, shows their application to two boxes and *B*, to one. Collars should be located by the side of such boxes as are well braced, otherwise the hanger will vibrate and the belts will be set to swinging.

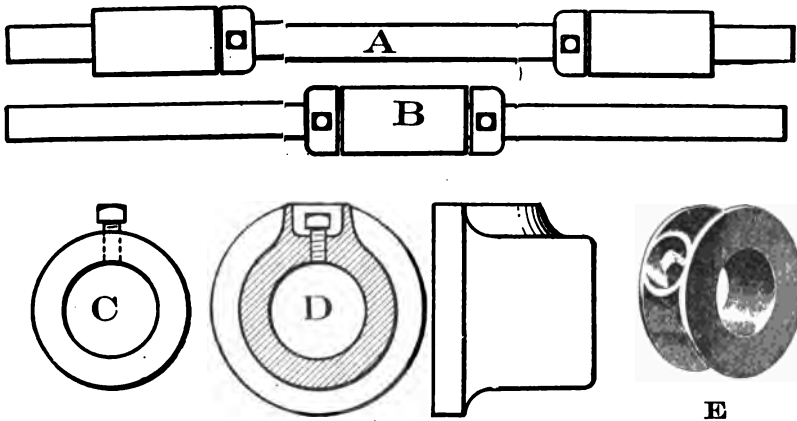


FIG. 56.

The simplest set collar is shown in *C*. This should not be used unless properly protected, because of the danger of the workman being caught with the screw. *Safety collars* as *D* and *E* should always be used.

Split Collars are made of the form similar to *E*. These can be removed or applied to the shaft without removing the shafting from the box.

134. References for Hubs, Keys, and Couplings:—Low and Bevis, "Machine Drawing and Design," pages 65-72; Reuleaux, "The Constructor," pages 95-101.

Toothed Gearing.

135. Gear Teeth:—Gear teeth are formed in three ways *i. e.*, pattern and machine molded, and machine cut. For large, slow moving or rough work the pattern molded tooth is used; for a somewhat better grade of work where the velocity is not too high and where an accurate mesh is not required the machine molded tooth is used; while for high grade machines requiring high velocities or a good fit between the teeth, cut gears are required. Table XIV gives the ordinary proportions in use for the three kinds of teeth.

TABLE XIV.

p = Circular Pitch.

m = Diametral Pitch.

Kind of Tooth.	Addendum or Face.	Dedendum or Flank.	Clearance.	Working Height.	Width	Space
Pattern Molded.	.32 p	.38 p	.06 p	.7 p	.47 p	.53 p.
Machine Molded.	.32 p	.38 p	.06 p	.7 p	.48 p	.52 p.
Standard Machine Cut, or Long Tooth	.3183 p	.3581 p	.04 p	.6764 p	.5 p	.5 p
	$\frac{1}{m}$	$\frac{1}{m}$	$\frac{1}{8m}$	$\frac{2}{m}$.5 p	.5 p
Machine Cut, Short Tooth	.25 p	.3 p	.05 p	.55 p	.5 p	.5 p

The two forms of gear teeth in general use are the *epicycloid* and the *involute*. The latter is preferred by many manufacturers. The following sizes for the involute tooth are used by the respective companies.

Brown Hoisting and Conveying Co.

Addendum = A ; dedendum = B ; radius at root of thread = R ; width of tooth = T ; m = diametral pitch, and p = circular pitch.

$$\text{Cast tooth, } A = \frac{.75}{m} \quad B = \frac{1}{m} \quad R = B - A \quad T = .46 p.$$

$$\text{Cut tooth, } A = \frac{.75}{m} \quad A + B = \frac{1.657}{m} \quad R = \frac{.157}{m}$$

Wellman, Seaver, Morgan Co.

$A = .2 p$; $B = .2 p + \text{clearance}$; $T = .47 p$ (cast) and $.5 p - \frac{1}{8}$ (cut); clearance = $.05 (p + 1)$ (cast), and $.03 (p + 1)$ (cut). Angle of action = 15° .

See also, Stahl and Wood's, Mechanism, Page 104.

TABLE XV.

Standard Pitches of Gear Teeth.

Pitch.		Pitch.	
Standard Diametral <i>m</i>	Corresponding Circular <i>p</i> , inches	Standard Circular <i>p</i> , inches	Corresponding Diametral <i>m</i>
$\frac{1}{2}$	6.283	$4\frac{1}{2}$	0.698
$\frac{3}{4}$	4.189	4	0.783
1	3.142	$3\frac{1}{2}$	0.898
$1\frac{1}{4}$	2.513	3	1.047
$1\frac{1}{2}$	2.094	$2\frac{3}{4}$	1.142
$1\frac{3}{4}$	1.795	$2\frac{1}{2}$	1.257
2	1.571	$2\frac{1}{4}$	1.396
$2\frac{1}{4}$	1.396	2	1.571
$2\frac{1}{2}$	1.257	$1\frac{7}{8}$	1.676
$2\frac{3}{4}$	1.142	$1\frac{3}{4}$	1.795
3	1.047	$1\frac{5}{8}$	1.933
$3\frac{1}{2}$	0.898	$1\frac{1}{2}$	2.094
4	0.785	$1\frac{1}{8}$	2.135
5	0.628	$1\frac{3}{8}$	2.285
6	0.524	$1\frac{5}{16}$	2.394
7	0.449	$1\frac{1}{4}$	2.513
8	0.393	$1\frac{3}{16}$	2.646
9	0.349	$1\frac{1}{8}$	2.793
10	0.314	$1\frac{1}{16}$	2.957
11	0.286	1	3.142
12	0.262	$\frac{15}{16}$	3.351
14	0.224	$\frac{7}{8}$	3.590
15	0.209	$\frac{13}{16}$	3.867
16	0.196	$\frac{3}{4}$	4.189
18	0.175	$\frac{11}{16}$	4.570
20	0.157	$\frac{5}{8}$	5.027
22	0.143	$\frac{9}{16}$	5.585
24	0.131	$\frac{1}{2}$	6.283
26	0.121	$\frac{7}{16}$	7.181
28	0.112	$\frac{3}{8}$	8.378
30	0.105	$\frac{5}{16}$	10.053
32	0.098	$\frac{1}{4}$	12.566
36	0.087	$\frac{3}{16}$	16.755
40	0.079	$\frac{1}{8}$	25.133
48	0.065	$\frac{1}{16}$	50.266

The rough cast tooth is made stronger than the machine cut tooth because at times the action of the load may be on the corner of the tooth rather than along its whole face. This may be due to the poorly shaped teeth in the pattern or to poor alignment in the machine.

136. Pitch of Spur Gear Teeth:—The force exerted on a tooth is continually changing in value and its line of action relative to the radius of the wheel; this makes the calculation of the gear tooth a rather unsatisfactory problem. In the following it will be assumed that, working under usual conditions, the maximum load on the tooth will act at the end of the tooth and that its value will be equal to the total load on the gear, (see note under formula 40). It will be assumed that the tooth has a rectangular section throughout.

Not one of the above assumptions is exactly correct but each is accepted as being the best approximation that can be made. In explanation of the above it may be said: First, that the maximum load, W , on the end of the tooth acts in a line that is not perpendicular to the line joining the centers of the gears, although the friction between the teeth has a tendency to bring it perpendicular, how nearly this is true in practice is not an easy matter to determine, hence the maximum value is taken. Second, the maximum load on the tooth is not known to be the full value of the turning force, for instance, if the arc of action for a pair of gears in contact is twice the pitch, the condition usually existing, there would be two pairs of teeth in contact and the pressure, W , on one tooth, would be approximately $\frac{P}{2}$, but, since, because of poor alignment, springing

shafts, or irregular shaped teeth the full load *may* come on one tooth only, it is best, in ordinary practice with usual conditions, to design each tooth to take the full load, $P = W$. Third, the tooth outline is not rectangular, but varies from this on account of the *kind, number* and *construction* of the teeth, diameter of gear, etc. In some teeth, especially those on large wheels the outline will be much stronger,

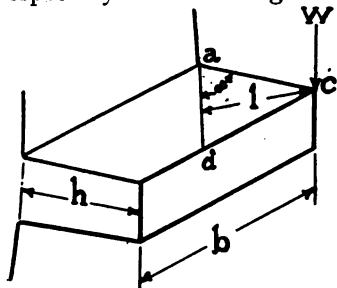


FIG. 57.

while for small gears it may be much weaker than the rectangle. As an average section the rectangle is accepted.

Suppose the worst condition to be when the load, W , acts at the corner of the tooth Fig. 57, in which case the tooth will break along some line as $ad = x$, and we have

$$Wl = \frac{f x t^2}{6} \quad (37)$$

Assume both W and t constant then f is a maximum when $\frac{l}{x}$ is a maximum. Since $l = h \sin \theta$ and $x = \frac{h}{\cos \theta}$, $\frac{l}{x} = \sin \theta \cos \theta$, which is a maximum at 45 degrees and gives $\frac{l}{x} = \frac{1}{2}$

Substituting these values $\frac{l}{x} = \frac{1}{2}$, and $t = 0.47 p$, from Table XIV, into (37) we have the circular pitch,

$$p = 3.68 \sqrt{\frac{W}{f}} \quad (38)$$

For rough gears $b = 1.5 p$ to $2 p$.

In modern practice the diametral pitch, m , i. e., the number of teeth per inch of pitch diameter, is largely used in describing gear wheels. For instance a 4 pitch gear is one having 4 teeth per inch of diameter. On page 103, (Table XV) are given the standard diametral and circular pitches in general use, and, in so far as practicable, these values should be adhered to in the design of gear teeth, since deviation from them would make special patterns or gear cutters necessary.

Next, considering the force, W , acting along the entire face of the tooth and perpendicular to it

$$W h = \frac{F b t^2}{6} \quad (39)$$

from which, if $t = 0.47 p$, as before, $h = 0.7 p$ and $b = n p$, where $n = 3$,

$$p = 2.5 \sqrt{\frac{W}{f}} \quad (40)$$

Note:—This formula (40), and those which follow, (41), (42), (43) and (44), developed from the same source, properly apply only to small pinions where the total turning effort, P , is considered as acting on one tooth, but as the error would be on the side of safety, it can be used for moderate sized wheels where great accuracy is not required. For large gears W should be taken as $\frac{P}{2}$, owing to the increased strength of section due to tooth outline, as mentioned above.

The usual rule for standard machine cut teeth is to make $t = \frac{F}{2}$ allowing no calculable back-lash, to make $h = \frac{1}{8}$ $m = 0.6764 p$ and $b = 3 p$. Substituting in (39) we have, for standard teeth,

$$0.6764 p W = \frac{3 p f x (5 p)^2}{6}$$

$$p = 2.33 \sqrt{\frac{W}{f}} \quad (41)$$

$$m = 0.741 \sqrt{\frac{W}{f}} \quad (42)$$

There is however a marked tendency at the present time towards the use of *short* teeth, since such forms show greater strength in service, less obliquity of action, and less expense in cutting. For short teeth, of the dimensions indicated in Table XIV, the corresponding formulas for the circular and diametral pitch are:

$$p = 2.1 \sqrt{\frac{W}{f}} \quad (43)$$

$$m = 0.67 \sqrt{\frac{W}{f}} \quad (44)$$

The value of n may vary between 2 and 3; the latter value being that which has been in general use. The tendency now however, is to decrease b and increase p , which would necessarily reduce the value of n .

137. Lewis' Formulas:—The foregoing formulas can only be regarded as approximate since the strength of gear teeth depends upon the number of teeth in the wheel. In teeth of both the involute and cycloidal forms, there is a marked difference between racks and pinions in working strength, since the teeth of a rack are broader at the base and consequently stronger than those of a pinion, while in radial flanked teeth, which are used more especially on bevel gears, the difference is not so pronounced. Mr. Wilfred Lewis reported, in the Proceedings of the Engineers' Club of Philadelphia, Vol. X, 1893, page 16; also in the American Machinist, May 4, 1893, page 3, certain formulas, for the design of cut spur and bevel gears, which he had deduced to take into account this variation, in strength, due to number of teeth. The scheme for the gear tooth design which he outlined has met with such universal favor and wide spread usage since its introduction that it may almost be referred to as the accepted standard method of design. Mr. Lewis in his investigation assumed the total turning effort to be

taken by one tooth and the force, $\dot{W} = P$, as being effective at the point in the tooth where the normal to the tooth curve at its upper point intersects the median line of the tooth. At this point the pressure between the teeth is resolved into two components—one perpendicular to the radius of the gear through the point, and the other radial. This radial component is neglected, only the tangential force which exerts a purely bending action on the tooth, being considered. In accordance with these assumptions, substitution was made in formula (39), yielding the expression

$$W = p f b \frac{2x}{3p}$$

We need not be concerned with the value of x further than to say that $\frac{2x}{3p} = y$. This is the factor of strength, varying with the number of teeth; it is determined by graphical construction, and is given in Table XVI, for reference.

$$\text{Then} \quad p = \frac{W}{fby} \quad (45)$$

or if $b = 3p$ as in formula (41), then

$$p = .578 \sqrt{\frac{W}{fy}}$$

TABLE XVI.

FACTOR FOR STRENGTH, y .			
NUMBER OF TEETH	Involute 20° Obliquity.	Involute 15° and Cycloidal.	Radial Flanks.
12	.078	.067	.052
13	.083	.070	.053
14	.088	.072	.054
15	.092	.075	.055
16	.094	.077	.056
17	.096	.080	.057
18	.098	.083	.058
19	.100	.087	.059
20	.102	.090	.060
21	.104	.092	.061
23	.106	.094	.062
25	.108	.097	.063
27	.111	.100	.064
30	.114	.102	.065
34	.118	.104	.066
38	.122	.107	.067
43	.126	.110	.068
50	.130	.113	.069
60	.134	.114	.070
75	.138	.116	.071
100	.142	.118	.072
150	.146	.120	.073
300	.150	.123	.074
Rack.	.154	.124	.075

It should be noticed in the application of formula (45) that either the *pinion* or the *wheel*, gearing together, may be the measure of strength of the combination; it therefore becomes necessary to investigate in each case, for the gear with the weaker form of tooth must under all conditions determine the pitch to be used.

To illustrate:—Let it be required to find the working strength of a 12-toothed pinion of 1 inch circular pitch, $2\frac{1}{2}$ inch face, driving a wheel of 60 teeth at 100 feet or less per minute, and let the teeth be of the 20-degree involute form. For a cast-iron pinion we have,

$$bp = 2.5, \quad y = .078, \text{ and } f = 8000 \quad (\text{Table XVII}).$$

Multiplying these values together we have $W = 1560$ pounds. For the wheel $y = .134$, and $W = 2680$ pounds. The cast-iron pinion is, therefore, the measure of strength, but if a steel pinion be substituted, we have $f = 20000$, and $W = 3900$ pounds, in which combination the wheel becomes the measure of strength, since it is the weaker.

For standard machine cut spur gears, cycloidal system or involute system, with the angle of obliquity 15 degrees, the Lewis formulas are as follows:

$$W = n p^2 f \left(0.124 - \frac{0.888}{N} \right) \quad (46)$$

where $N =$ the number of teeth.

$$p = \sqrt{\frac{W N}{n f (0.124 N - 0.888)}} \quad (47)$$

In the application of all the "Lewis" formulas given above it becomes necessary to know, N , the number of teeth. In the determination of the pitch by any method the diameter of the gear must be known or assumed in order to determine the tangential turning force, P , which produces the stress in the teeth. The relation of the number of teeth to the diameter is such that when they are given the pitch is determined. This being the case the best method of procedure in applying the above formulas would be to assume some standard value of m as given in Table XV, since in any case one of these values must be accepted as the result. This assumed value of m with the pitch diameter determines N , and we may solve for W , the safe load for the tooth. A comparison of this value with the load which the gear is required to transmit will show whether or not the assumption for m was correct. Whatever error is permitted should be on the side of safety, but if some degree of judgment is shown in selecting m a second or, at most, a third trial should give practical working results, although the process might be continued to any desired limit of exactness. An alternative method of procedure in designing the teeth of gears where the number is unknown, would be to obtain the dimensions approximately with formula (41) or (43), and then correct these values by using one of Lewis' formulas.

have given good results for a number of years in the design work of certain manufacturing companies; and their correctness has been closely verified by Prof. Benjamin in a series of experiments on the breaking strength of gear teeth. While it cannot be claimed that these values should be taken for general adoption, their deviations, such as they may be, are believed to be in the right direction, for safety and durability.

These fibre stresses will not agree entirely with those used in ordinary machine construction, but experiments conducted under conditions of service show the ultimate strength of gear teeth to differ quite materially from the nominal ultimate strength of the material, thus verifying these special values, which should not be applied to other than gear work.

Reuleaux gives the following formula for cast iron transmission gears, in which the linear velocity, V is more than 100 feet per minute.

$$f = \frac{960000}{V + 2164} \quad (53)$$

For steel f may be taken 3.33 times, and for wood or fibre 0.6 times the value thus obtained.

139. Pitch of Bevel Gear Teeth:—To determine the pitch diameter, pitch, number and length of teeth of bevel gears the following formula, due to Lewis, may be generally applied and gives fair satisfaction.

$$W = f p b' y \frac{D^3 - d^3}{3 D^2 (D - d)} \quad (54)$$

where f , p , and W are the same as given in Pars. 136, 137 and 138; D and d are the largest and smallest pitch diameters, respectively, of the frustrum of the pitch cone; b' is the breadth of face of the bevel gear tooth, and y is taken from Table XVII to correspond with N , the "equivalent" number of teeth, *i. e.* the actual number of teeth on the bevel wheel x secant a . Where a = slope of cone, or half angle of pitch cone.

In most cases d should not be less than $\frac{1}{3} D$ and (54) will give good results if taken

$$W = f p b' y \frac{d}{D} \quad (55)$$

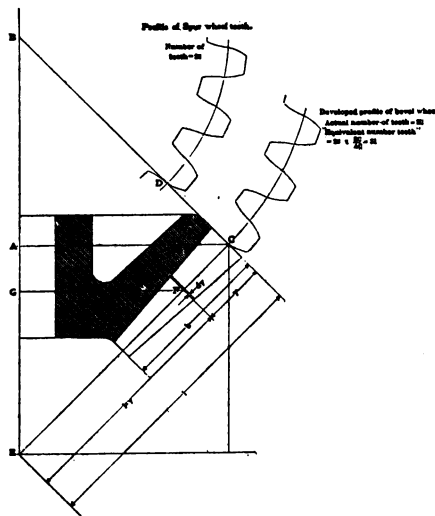
Note:—In applying (54) or (55) it will be found more convenient to assume the gear sizes and solve for the force W . If, however, W is given to find the size of the gear, proceed as follows: find p as in a spur gear; take f and y from the tables; assume D , and by construction find a relation between d or b' ; then by substitution either d or b' may be obtained.

Another method which will give fairly satisfactory results is to make the pitch and the length of the bevel gear tooth, equal to those

of a spur gear, whose pitch diameter of the frustrum of the pitch cone of the bevel wheel.

The following alternative equation may be derived directly from that given by Mr. Lewis, and is, in fact, the same equation expressed in different terms, the purpose being to bring bevel gears under the same rules as those for finding the strength of spur gears.

In considering the strength of bevel wheels an examination of a bevel gear drawing is sufficient to show that the formula for spur wheels needs modification, not only on account of the tapering section of the teeth, but also because the tooth profile is itself considerably modified by the shape of the tooth, being no longer solely dependent upon the number of teeth in the gear, but being also influenced by the angle of the pitch cone. In the case of the bevel wheel, the teeth are drawn on the development of a cone whose slant height is BC , Fig. 58, the effect being that the tooth is strengthened considerably, owing to the fact that in laying out the tooth profiles, BC , is used as the pitch circle radius instead of BD . The tooth



Showing Increased Strength of Bevel Teeth as Compared with Spur Teeth of Same Number and Pitch.

FIG. 58.

profile is, in fact, that of a wheel having the same pitch, but the number of whose teeth is increased in the ratio of BC to BD . It is, moreover, obvious that as regards strength, the bevel wheel may be supposed to possess a greater *effective* number of teeth, in a measure depending upon the angle of the pitch cone. In the case considered since the angle BCA is equal to the half angle of pitch cone, vis.,

AEC, it can be seen that the number of effective teeth can be found by multiplying the *actual* number of teeth by the secant of the half angle of pitch cone.

The exact amount by which the tapering section of tooth reduces the safe load which it can carry can be determined if the ratio, r , of the breadth of face, b' , to l , the length of the slant height of the pitch cone, is known. This ratio should be taken as 1 to 3 where possible.

If these facts are kept in mind, bevel wheels may be designated by using any of the Lewis formulas for spur gears, provided the "equivalent" number of teeth and the corresponding value of y , are employed instead of the actual values of these terms; although the speed and diameter should be taken the same as those of the pitch line of the bevel gear. In other words, the strength of the bevel wheel tooth will be the same as that of a tooth of a spur-wheel, carrying a number of teeth which bears the proportion of BC to AC to the number of teeth in the bevel gear. Then, if N' , the actual number of teeth in the bevel gear, be multiplied by secant a , *i. e.* the secant of the half angle of the pitch cone, the resultant number may be taken as N , the "equivalent number of teeth" to be used in designing the spur wheel.

Now the safe load for the bevel gear will be the same as that for this equivalent spur gear, provided that, instead of using the simple breadth, b' , of the face of the bevel wheel a quantity, b , less than it (in a manner depending upon the value of r ; and which may be conveniently called the "equivalent breadth") be used in calculating the spur wheel.

By a process of integration this equivalent breadth has been found to be,

$$b = b' \left(1 - r + \frac{r^2}{3} \right) \quad (56)$$

The calculation of the "equivalent breadth" may be avoided by the use of the diagram Fig. 59, in the following manner: Find r by dividing the desired length of face of the bevel wheel by the length of the slant height of the pitch cone. On the lower scale of the diagram find the value of this ratio, and from the point so found

trace vertically till the curve marked $\left(1 - r + \frac{r^2}{3} \right)$ is intersected.

From this point of intersection trace a horizontal line to intersect the diagonal marked with the breadth of face of the bevel wheel in inches, thence follow vertically till the value of the equivalent breadth is read off on the top scale. If the breadth of face of the bevel gear is greater than the dimensions marked on the diagonals, divide the breadth by two, and having found the "equivalent breadth," multiply it by two. The full lines in Fig. 59, show how the equivalent breadth is found, when $r = .25$ and the actual breadth of face = 5 inches.

With the value of b determined, if N' is known, the value of N for the equivalent spur gear may be found,

$$N = N' \sec a, \text{ as above;}$$

and substitution made in formulas (51) or (52); but this procedure will lead to the same difficulty pointed out in the discussion following formula (47), since both diameter and number of teeth are involved in these formulas for determining pitch.

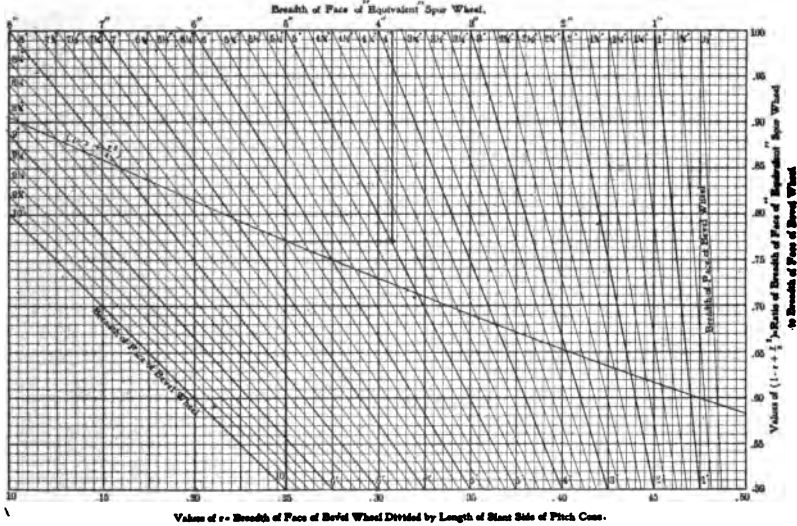


Diagram for Reducing Face of Bevel Gear to Equivalent Face on Spur Gear.

FIG. 59.

Directions:—Find r on Lower Scale and Trace Vertically to the Curve, thence Horizontally to Intersection with Diagonal Giving Face of Bevel Wheel, thence Vertically to Value of "Equivalent" Face of Spur Wheel on Top Scale.

To avoid this difficulty, when the value of b has been ascertained, the following formulas,

$$W = b p f \left(0.124 - 0.282 \frac{p}{D \sec a} \right) \quad (57)$$

$$p = D \sec a \left(0.22 - \sqrt{0.048 - 3.55 \frac{W}{b f D \sec a}} \right) \quad (58)$$

will be found helpful. Formulas (57) and (58) are deduced from (49) and (50), so as to take into account the equivalent number of teeth instead of the actual numbers; and by direct substitution in them the circular pitch common to the equivalent spur and bevel wheel will be determined.

Example:—Required a set of mitre gears, of approximately 18 inches pitch diameter, to transmit 25 horse power at 150 revolutions per minute.

Since the pitch radius for each gear is 9 inches, $l = 13.856$ inches. Let $r = \frac{1}{3}$, then $b' = 4.618$, say $4\frac{5}{8}$ inches; substituting in formula (56) or reading directly from diagram, Fig. 59, $b = 3.4$

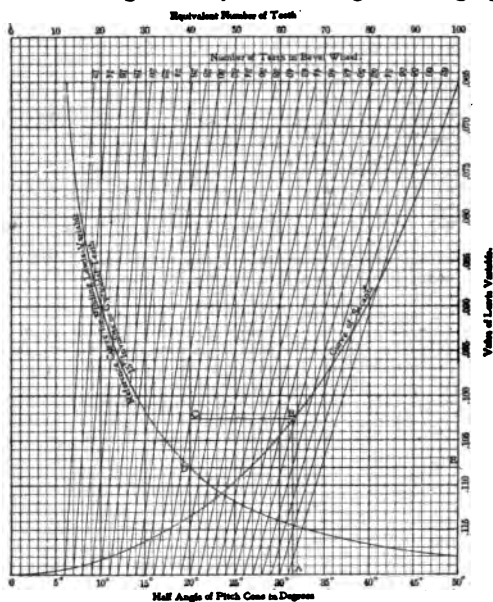


Diagram for Reducing No. of Teeth of Bevel Gear to Equivalent No. of Teeth of Spur Gear

FIG. 60.

inches. Velocity = 707 feet per minute; $W = 1169$ pounds; then, interpolating in Table XVII, $f = 3680$ pounds; and solving in formula (58)

$$p = 18 \times 1.414 \left(0.22 - \sqrt{0.048 - \frac{3.55 \times 1169}{3.4 \times 3680 \times 18 \times 1.414}} \right) = 0.84 \text{ inches.}$$

Referring to Table XV the next larger standard circular pitch is $\frac{7}{8}$ inch, then

$$N = \frac{18 \times 3.1416 \times 8}{7} = 65.6, \text{ say } 66 \text{ teeth, which gives the}$$

exact pitch diameter of the bevel wheel as,

$$\frac{7 \times 66}{8 \times 3.1416} = 18.38 \text{ inches.}$$

140. Gear Arms and Rims:—The size of the arm at the center of the wheel may be figured by moments, where $f = 2500$ pounds per square inch; the load on the arm $= W \div$ No. of arms considered as carrying the load; and $l =$ radius of gear. Having the section at the center, that at the rim is given the usual proportion.





$b = 3.5p - \sqrt{\frac{P}{f}}$		$p = \text{Pitch}$			
		A	B	C	D
	Sec. at	$2p + .06l$	$p + .03l$	$\frac{2}{3}A$	
	Sec. at	$2p$	p	$\frac{2}{3}A$	
	Sec. at	$2.25p + .06l$	$.5p$	$b - 1.6B$	$.3p$
	Sec. at	$2.25p$	$.5p$	$b - 2B$	$.3p$
	Sec. at	$2.4p + .06l$	$.5p$	$b - .5p$	$.3p$
	Sec. at	$2.4p$	$.5p$	$b - .5p$	$.3p$
	Sec. at	$2.3p + .06l$	$.5p$	$b - 1.5B$	$.3p$
	Sec. at	$2.3p$	$.5p$	$b - 2B$	$.3p$

FIG. 61.

Fig. 61 gives approximate sizes for sections that are sometimes recommended.

In planning for bevel gears it is customary to locate them either *against* or *near* to a journal box. It will be seen, B, Fig. 62, that

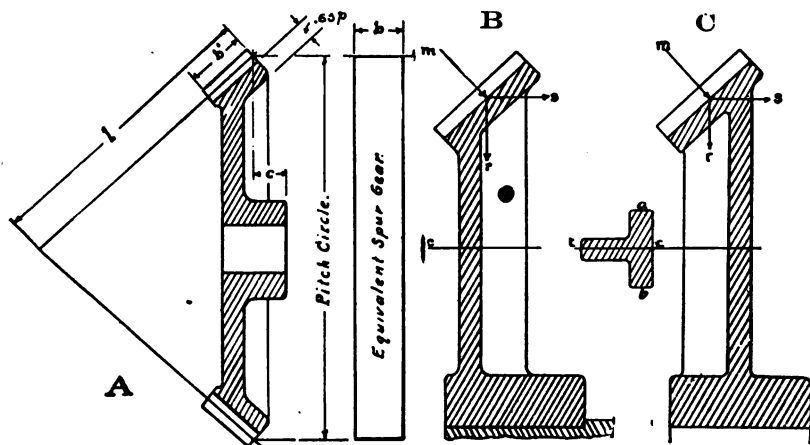


FIG. 62.

in addition to the torsion on the shaft due to the force P , there is a bending of the shaft due to the force s which is a resultant of the force m tending to separate the gears. It is therefore evident that the nearer the gear is to the support, the less the bending of the shaft.

The rims of small gears are connected to the hubs by a solid disc as in A . In larger gears arms are made as in B and C . The T shaped arm is very common in gear work. The disc part of the arm ab is figured to stand the entire turning moment and the web is added to resist the side thrust s . B is stronger to resist the bending than C , but the latter is more commonly used because of its smooth back which adds safety in operation. It is undoubtedly true also that the rim in C is better supported than in B .

In fitting a shaft to a bevel gear it is usually *shouldered* as shown in B .

The thickness of the rim under the tooth may be taken .65 p and the backing c may be taken for ordinary conditions at

$$c = \begin{cases} \frac{1}{8} D + \frac{1}{4}'' & \text{for gears below 24'' diameter.} \\ \frac{1}{8} D - 1'' & \text{for gears above 24'' diameter.} \end{cases}$$

141. References for Toothed Gearing:—Lewis, "Investigation of the Strength of Gear Teeth," American Machinist, May 4th, 1893, page 3; Jones, "Machine Design," Part II, pages 75-104; Benjamin, "Machine Design," pages 146-156; Reuleaux, "The Constructor," pages 144-150; Low and Bevis, "Machine Drawing and Design," pages 173-191; Grant, "Gearing;" Brown and Sharp, "Formulas in Gearing" and "Practical Treatise on Gearing;" Stahl and Woods, "Elementary Mechanism," pages 66-130.

142. Problems.—

(1). Calculate, by Lewis' method, the diametral pitch and number of teeth of a standard cut cast iron gear 12 inches in diameter, running at 250 revolutions per minute and transmitting 30 horse power.

(2). A standard machine cut cast iron gear wheel is 9 feet $6\frac{1}{2}$ inches in pitch diameter, and has 120 teeth with width of face of 9 inches. Determine the circular and diametral pitches of the teeth, and the horse power which the gear will transmit safely when making 80 revolutions per minute.

(3). A $1\frac{1}{2}$ pitch cycloidal tooth, 6 inches broad, in a wheel of 86 teeth, failed under a load of 42000 pounds. Find f , considered as an ultimate value, by Lewis' formula.

(4). The drum of a hoist is 16 inches in diameter and makes $2\frac{1}{2}$ revolutions per minute. The diameter of the gear on the drum is 36 inches and of its pinion 6 inches. The gear on the counter shaft is 24 inches in diameter and its pinion is 6 inches in diameter. The gears are all machine cut cast steel. The load on the drum chain is 2 tons.

Determine the horse-power of the machine, and calculate the pitch and number of teeth of each gear; (a) with standard teeth, (b) with short teeth instead of standard.

(5). A vertical water-wheel shaft is connected to a horizontal drive shaft by standard machine cut cast iron gears. The water-wheel makes 200 revolutions per minute and yields 225 horse power. The drive shaft runs at 160 revolutions per minute. Determine the dimensions of the gears if the smaller is approximately 3 feet 6 inches in pitch diameter.

Friction Gearing.

143. Friction Gearing:—Friction wheels are used in light power work where the service is intermittent, where the velocity ratio of the wheels is changeable, and where high speeds would cause toothed wheels to be noisy. They are generally used to connect shafts that are parallel or at 90 degrees with each other, but may be used to connect shafts at any angle.

The *materials* used in friction wheels are iron, wood, paper or mill-board, and leather. The driver should be made from the softer material and the follower from the harder. This is a protection against wearing the face of the follower unevenly in case of slipping. The usual materials employed are, paper for the driver and iron for the follower.

The relation existing between the pressure on the wheels perpendicular to the surface at the line of contact and the rotative force transmitted may be shown for the different conditions as follows:

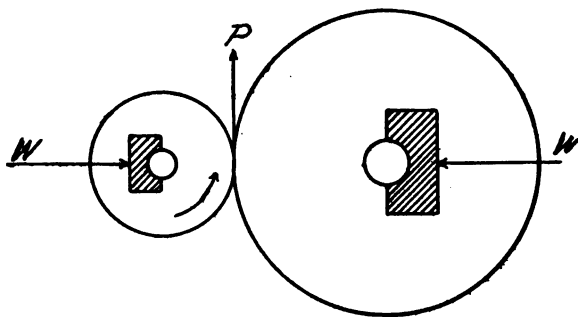


FIG. 63.

For *parallel shafts*, Fig. 63, let W = pressure in pounds, total, w = pressure in pounds per lineal inch of wheel face b , P = total rotative force in pounds and Φ = coefficient of friction between the two surfaces; then $W \Phi = P$, or $b w \Phi = P$. Substituting these

values in the standard horse power formula we have, if V = velocity of the rim in feet per minute.

$$H. P. = \frac{W \Phi V}{33000} = \frac{b w \Phi V}{33000} \quad (59)$$

This formula may be used in determining the power transmitted by any given set of wheels, or to determine the required pressures between the surfaces of any two given wheels transmitting a given power. The coefficient of friction Φ , may be taken according to Unwin, page 283, as follows: Metal to metal, .15 to .2; paper to metal .2; and wood to metal .25 to .3.

For *shafts at 90 degrees*, Fig. 64, let W be the pressure normal to the surface as before. Resolve this into forces R_1 , S_1 , R_2 and S_2 , parallel and perpendicular to the respective shafts. Then

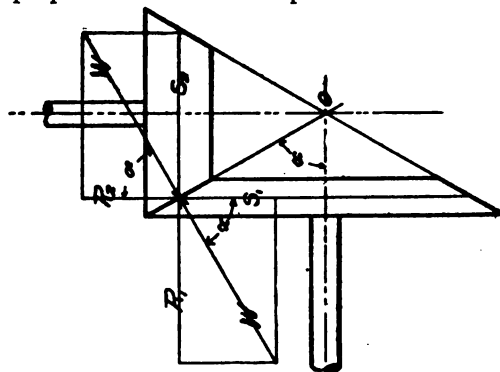


FIG. 64.

$$R_1 = W \sin a = S_2$$

$$R_2 = W \cos a = S_1$$

substitute these in the standard horse power formula and obtain

$$H. P. = \left\{ \begin{array}{l} \frac{R_1 \Phi V}{33000 \sin a} = \frac{S_2 \Phi V}{33000 \sin a} \\ \frac{R_2 \Phi V}{33000 \cos a} = \frac{S_1 \Phi V}{33000 \cos a} \end{array} \right. \quad (60)$$

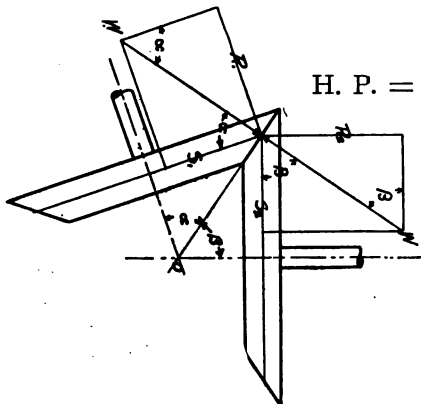


FIG. 65.

For *shafts at any angle* Fig. 65, let the notation be as before and the following formulas will be true:

$$R_1 = W \sin a.$$

$$S_1 = W \cos a.$$

$$R_2 = W \sin B.$$

$$S_2 = W \cos B.$$

then by substitution

$$H. P. = \frac{R_1 \Phi V}{33000 \sin a} = \frac{S_1 \Phi V}{33000 \cos a} = \frac{R_2 \Phi V}{33000 \sin B} = \frac{S_2 \Phi V}{33000 \cos B} \quad (61)$$

144. Width of Face:—To determine the width of the face of a friction wheel, use the latter part of formula (59.) Unwin quotes the values $w \Phi$ as, 30 pounds for maple wood, 15 to 20 pounds for pine wood and 80 pounds for paper. The statement is also made that the width of face of the friction wheel may be taken the same as the width of a single leather belt transmitting the same power at the same velocity

145. Pressure of Contact, Coefficient of Friction and Horse Power:—In the use of friction wheels, two factors enter which are more or less uncertain and difficult to determine. One, the coefficient of friction, has just been mentioned, and the other is the slippage between the two wheels.

Probably no values may be quoted for them with more assurance than those relating to paper and iron contact, although values that are considered very safe have been determined for other materials as well. Experiments conducted in the laboratory of Purdue University under the direction of Dr. W. F. M. Goss and summarized in two papers before the American Society of Mechanical Engineers, one in December, 1896 and the other in December, 1907, are usually quoted as standard authority. The first set of experiments involved paper friction wheels of $5\frac{1}{2}$, 8, 12 and 16 inches diameter, in contact with a 16 inch cast iron wheel. The contact pressure varied from 75 to 400 pounds per inch of width, and the speed limits gave a peripheral velocity varying from 450 to 2700 feet per minute. More than 5000 observations were made. In the second set of experiments the following materials were tested for driving members: Straw fibre, straw fibre with belt dressing, leather fibre, leather, leather faced iron, tarred fibre and sulphite board. Each driving member was operated in combination with driven members of cast iron, aluminum, and type-metal. The results of the series showed that driving wheels of sulphite board and leather-faced iron, in connection with driven wheels of type metal were unable to withstand the severe conditions of service. Driving wheels of straw fibre with belt dressing possessed such small friction qualities as to be totally unfit for use. The other materials, however, proved quite satisfactory, and results obtained are included in the following deductions:

Pressure of contact:—With a constant coefficient of friction, the power transmitted varied directly with the pressure of contact. It is therefore desirable that the pressure be made as great as possible, to transmit the maximum power. On the other hand, excessive pressures served to break down the fibres of the material, and shortened the life of the wheel. It would therefore appear that the

life of any wheel depends, to a great extent, upon the pressure under which it operates. In the first set of experiments the paper friction wheels gave no signs of breaking down under pressures as high as 400 lbs. per inch in width. The safe pressures of contact per one inch of face that are recommended from these experiments are:

Straw fibre, 150 pounds.

Leather fibre, 300 pounds.

Leather 150 pounds.

Tarred fibre 250 pounds.

Concerning the effect of the pressure upon the driven wheels, it was found that aluminum was not as durable as cast iron under the high contact pressures admissible for tarred and leather fibre drivers.

Coefficient of friction:—The coefficient of friction was practically independent of the pressure of contact. It approached a maximum when the slip between the driving and driven wheels approached two per cent. It seemed to be more affected by the slip than by any other factor. By gradually increasing the load to be carried it was found that the slippage could be increased to three per cent, and, under favorable conditions, to four or even five per cent. When slippage increased above three per cent, results were not very satisfactory.

The following safe values for the coefficient of friction are recommended:

Straw fibre and cast iron.....	.255
Straw fibre and aluminum.....	.273
Leather fibre and cast iron.....	.309
Leather fibre and aluminum.....	.297
Leather and cast iron.....	.135
Leather and aluminum.....	.216
Tarred fibre and cast iron.....	.277
Tarred fibre and aluminum.....	.310

Horsepower:—The following general formula may be used for obtaining the horsepower of wheels in friction contact.

$$\text{H. P.} = \frac{\pi D N w b \phi}{33000 \times 12} \quad (62)$$

where D = diam. of friction wheel in inches.

N = no. of revolutions per minute.

b = width of face in inches.

w = pressure in lbs. per inch of face.

ϕ = coefficient of friction.

By substituting in the above formula, the horsepower for any condition of service may be obtained.

146. Forms of Friction Wheels:—Another form of friction gearing is the *disc friction* shown in Fig. 66 *A*, for shafts at 90 degrees, or the *cone friction* shown in *B*, for shafts at any angle. These are used to obtain variable speeds. It is evident that if the rotative speed of the driver be constant, the corresponding speed of the follower will depend upon the position of the driver relative to the center of the follower. Such gears are used on sensitive drills and other similar light machinery. In disc or cone frictions, a certain slippage takes place because it is not a pure rolling contact. This slippage can not be entirely eliminated.

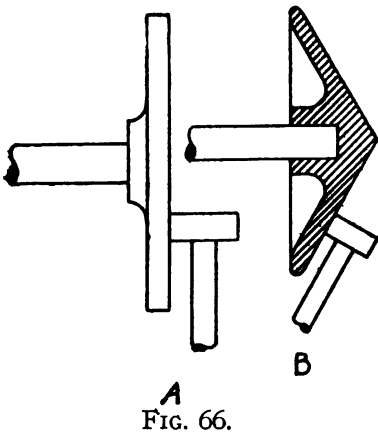


FIG. 66.

Fig. 67 shows a *reversible friction* which may be used on machines requiring slow forward and fast returning movements. With a constant speed of the driver, a slow forward speed may be obtained by forcing the driver against the inner surface of the outside rim, and to reverse, the driver may be forced against the outside surface of the hub of the wheel.

Wood friction wheels are constructed as shown in *A* and *B*, Fig. 68. Paper friction wheels are constructed as in *C*. The wood should be very uniform in texture, free from knots, and fine grained. Clear maple, pine or cottonwood may be used. The paper generally used is straw-board cut in large discs to fit closely to the shaft. These when compressed tightly between the flanges make a very dense and uniform material. In all flanges the bolt heads and nuts should be protected by ribs, or should be let into the flanges.

147. Cone Frictions:—Wedge or cone frictions are of two general classes, Fig. 69 *A*, two shafts in the same straight line, and *B*, two shafts in parallel. The first is used generally on power hammers and the second is used only on slow speed machinery because of the noise attending its use. The number of projections on *B* is between 1 and 6. Unwin gives the relation between R , the force holding the wheels together, and P the turning force on the shaft as

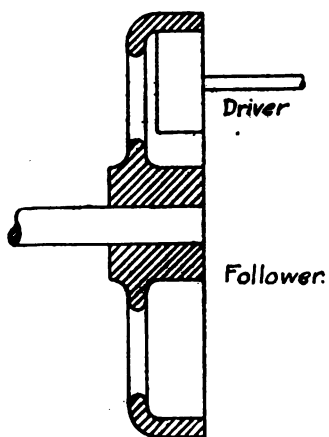


FIG. 67.

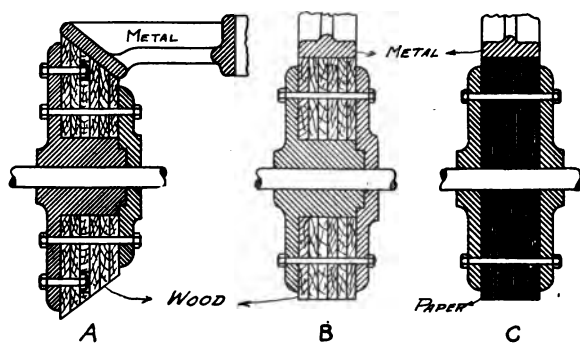


FIG. 68.

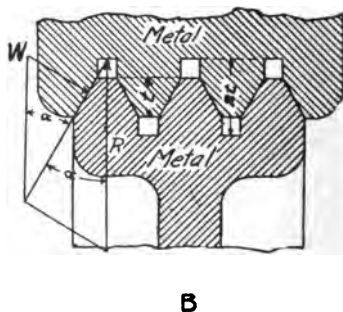
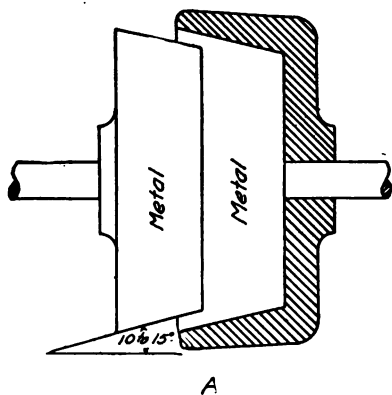


FIG. 69.

$$R = \frac{P}{\Phi} (\sin a + \Phi \cos a) \quad (63)$$

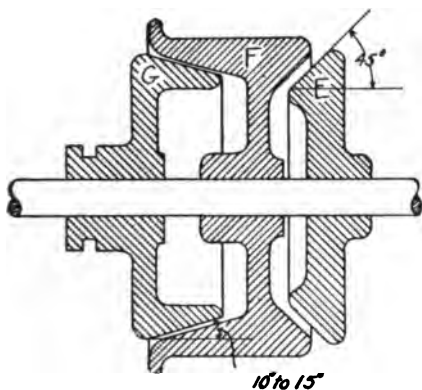


FIG. 70.

where Φ is the coefficient of friction and a is the angle of inclination between the side of the wedge and the vertical. This is usually taken between 15 and 20 degrees. The working depth of the tooth is

$$t = 0.025 \sqrt{P}$$

Fig. 70 shows a very effective form of friction gearing. In this, E is keyed fast to the shaft; G , fits over a sliding key and F is loose. When G is thrown to the right it wedges F between G and E and causes rotation of the shaft. This form gives a

more positive action than the single frictions previously mentioned.

148. References for Friction Gearing:—Jones, "Machine Design," Part II, pages 104-111, 235-242; Low and Bevis, "Machine Drawing and Design," pages 169-172; Reuleaux, "The Constructor," pages 122-126; The Rockwood Mfg. Co., Indianapolis, Ind., "Friction Transmission."

149. Problems:—

(1) A tarred fibre bevel friction wheel is to be 8 inches in diameter at the large end and is to have a working face of 8 inches. Under normal conditions of operation between shafts at right angles how many horse power will it successfully transmit at 325 revolutions per minute, when working with a cast iron follower? If the friction serves the purpose of decreasing speed and the velocity ratio is $4\frac{1}{2}$ to 1, find the small diameters of driver and follower, and the large diameter of follower. Find the half angle of the pitch cone for both driver and follower. How much adjustment along the axis will be necessary in order that the face of the fibre filler may be worn away $\frac{1}{4}$ inch, measured along the slant height of a cone normal to the working face?

(2) A paper (straw fibre) bevel friction running at 70 revolutions per minute is to be used to transmit 20 horse power with an increasing speed ratio of 1 to 5. The small diameter of the cast iron follower cannot be less than 6 inches. Determine the width of face, large diameter and pitch angles for both wheels. Determine the end thrust on each shaft. If an allowance of $\frac{1}{2}$ inch (measured

normal to the working face) is made for wear, how much adjustment along the shaft will be necessary?

(3). Same as Problem (2), but using leather fibre and cast iron instead of straw fibre and cast iron. Compare the results with those obtained for Problem (2).

Screw Gearing.

150. Worm Gearing:—In worm gearing Fig. 71, the worm is merely a long tooth wrapped around a cylinder; the pitch of the worm being the same as the pitch of the gear. This gearing is used in changing from a very high to a very low speed and is employed in the transmission of only small powers. The worm is the driver and, unless the pitch of the thread is very great, it will effectually lock the gearing. The wear on the worm is much greater than that on the gear.

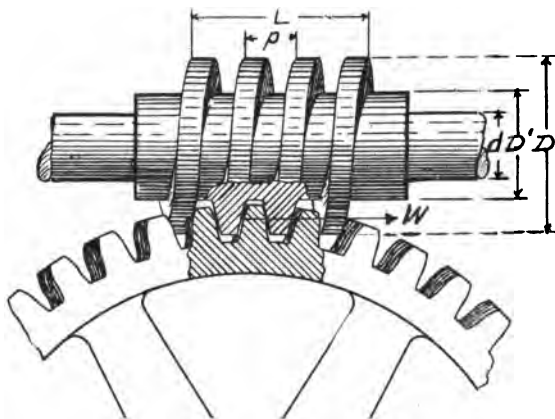


FIG. 71.

The action of the worm in connection with the gear is very much the same as a rack and spur gear. One revolution of the worm moves the gear through one circular pitch. The *ratio of the angular velocities* of the two shafts is the same as the *number of teeth on the gear*. It is advisable to make this ratio as large as possible. The minimum value is sometimes quoted as 30. When a smaller ratio is desired a *double thread* on the worm may be employed. A ratio very often used is 100, and gears are listed with a ratio as high as 250.

In designing the worm the following values will be found very good. Take the pitch p the same as that of a spur gear having the same W , then

$$D = 4 p$$

$$D' = 1.8 d \text{ to } 2 d.$$

$$L = 3 p \text{ to } 6 p, \text{ say } 4 p.$$

The teeth of the wheel may be straight and set at an angle from the center line of the shaft so as to conform with the slant of the worm thread, or they may be curved to fit the worm thread along the full length of the gear tooth. The latter method is preferred since, in the former, there is contact only along the center line of the gear.

Fig. 72 shows sections through the gear rim. *A*, is that of a straight tooth; *B*, is the ordinary form used on curved teeth, and *C*

is a modification of the curved tooth not so frequently used.

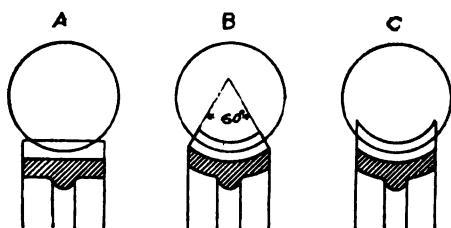


FIG. 72.

The tooth may be given the usual values for addendum, thickness, clearance, etc. The width of the face of the gear varies from $1.5 p$ to $2.5 p$. The involute tooth is preferred, and an involute wheel tooth working with a

screw tooth whose meridian section has straight, sloping sides, is the best combination, since such teeth may be cut with a simple hob, without the successive diminutions of the distance between the axes affecting the velocity ratio.

Right and left hand worms, Fig. 73, may be placed upon the same shaft, meshing into two gears that run in opposite directions. These gears operate two shafts, which in turn are connected by spur gears. The one shaft is an idler shaft, but serves to minimize the end thrust on the worm shaft.

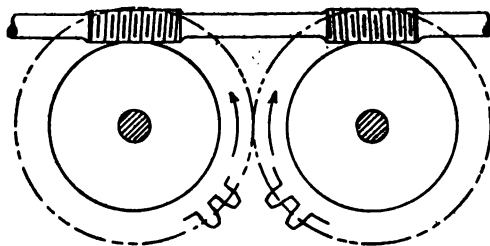


FIG. 73.

151. The Screw:—Most fastenings between machine parts are made through the action of the screw thread. The threads are classified as *square* and *V*, and the pieces thus threaded are classified as *bolts*, *studs*, *cap screws*, *set screws* and *machine screws*. See Par's 154 and 155, for standard sizes.

The action of the loaded screw, when it is turned about its own center, is equivalent to moving the same load along an inclined plane

whose angle with the horizontal is the same as the mean pitch angle of the thread.

152. Work Performed by Means of a Screw:—A very simple illustration of the overcoming of resistance by means of a screw is shown by Fig. 74.

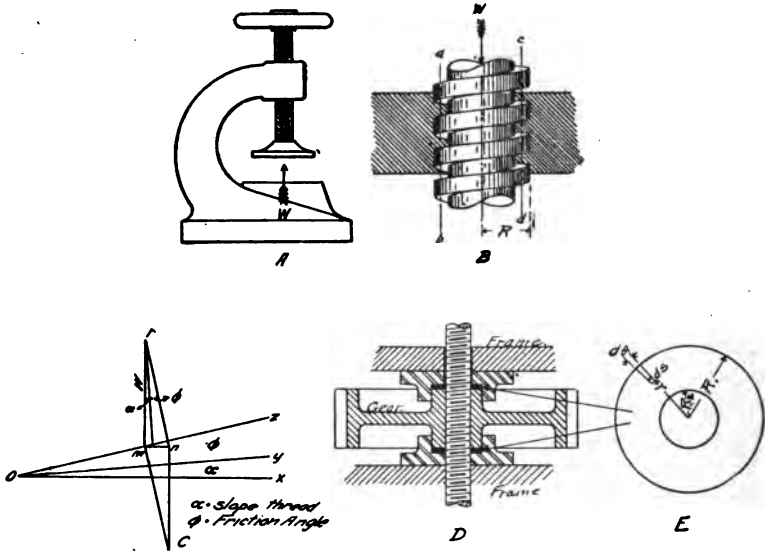


FIG. 74.

In *A* the square thread screw is sustaining a load of W pounds. Let this load be absorbed on a circle represented by the mean circumference of the thread whose diameter lies between ab and cd in *B*. Lay off this mean circumference ox in *C* and find the slope of the screw oy . Assume a coefficient of friction between the screw and the nut and lay off the line oz an angle Φ from oy .

With the force W perpendicular to ox , draw rn making the angle $(\alpha + \Phi)$ with W . Draw mn perpendicular to W , mn represents the turning effort P' in pounds, necessary to be applied at a radius R' from the center of the screw to move the load up the inclined plane against the action of its own weight and friction. One complete turn of the screw is equivalent, in effective work, to moving the load through the distance of one pitch.

Application 1. The main screw on a testing machine acts directly upon the test specimen. The maximum pull of the specimen is estimated at 15000 pounds. The screw is 2 pitch, is 2 inches mean diameter and is operated by a gear having 12 inches pitch diameter. What force will be exerted between the gear and the pinion in overcoming the load and the friction of the screw, if the latter coefficient be taken at .15?

The development of the screw thread gives an angle of $4^\circ - 30'$ and the angle represented by the coefficient of friction is $8^\circ - 30'$ making the total angle of 13° . From this is obtained

$$\tan. 13^\circ = mn \div W \text{ and } p' = mn = 3463 \text{ pounds.}$$

From moments we can readily obtain

$$\begin{aligned} 3463 \times 1 &= P R \\ \text{where } R &= \text{radius of gear} = 6'' \\ \text{and } P &= 3463 \div 6 = 577 \text{ pounds.} \end{aligned}$$

If instead of the gear, a two ended lever be used, the force exerted on each end of the lever would be $577 \div 2 = 288.5$ pounds.

Application 2. Given, the main screw of the above with the gear threaded to fit and with the end of the hub encased in an annular thrust bearing having 2.25 inches and 4.5 inches respectively, as the inner and outer diameters, Fig. 74 D. What would be the total force P on the gear necessary to overcome the load, the friction of the screw and the friction of the thrust bearing?

Considering first the thrust bearing, the turning moment necessary to overcome this friction is

$$P R = \frac{2}{3} \pi \Phi p (R_1^3 - R_2^3) \quad (64)$$

where P and R are the same as previously stated; Φ = coefficient of friction = .15; p = pressure per square inch of area on thrust bearing, and R_1 and R_2 are the radii respectively of the outer and the inner circumferences of the bearing.

Formula (64) represents the *frictional resistance offered by any annular disc*. *Proof.* Let $d s$ Fig. 74, E , be any increment of area at any radius r from the center of the disc. Then $d s = d r r d \theta$ and $r d s = r^2 d r d \theta$. From this is obtained

$$\begin{aligned} P R &= \int_{R_2}^{R_1} \int_0^{2\pi} p \Phi r^2 d r d \theta \text{ and} \\ P R &= \frac{2}{3} \pi \Phi p [R_1^3 - R_2^3], \text{ as given in} \end{aligned} \quad (64)$$

Substituting the above conditions in (64) we obtain

$$6 P = \frac{2 \times 3.14 \times .15 \times 1258}{3} [(2.25)^3 - (1.125)^3]$$

$P = 656.9$ pounds on the thrust bearing.

Then $577 + 656.9 = 1233.9$ pounds, total force at gear for load and frictions.

For the two ended lever this would be 616.9 pounds on each end of the lever.



The *total energy* exerted in one complete revolution of the screw is (not counting the thrust bearing) $577 \pi D = 1813.4$ foot pounds; and counting the friction of the thrust bearing it is $1233.9 \pi D = 3873$ foot pounds.

The *useful work* done is $15000 \times .5 = 7500$ inch pounds or 625 foot pounds for each revolution.

The *efficiency of the screw* (including the thrust bearing) is $625 \div 3875 = .16$; and without the thrust bearing it is $.34$. This latter efficiency may also be shown as $\tan a \div \tan (a + \phi)$.

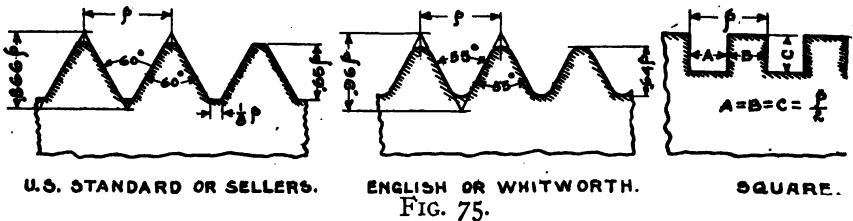
153. Strength of V-Thread:—Where screws are used for transmission purposes, the square thread is preferred. This choice is made largely because of the *wedging action* of the *V* thread, this having a tendency toward splitting the nut. This wedging force on a *V* thread may be obtained in any case if desired. Let W = force parallel with the center of the screw, a = the angle of the screw thread, and Q = force perpendicular to W ; then $Q = W \tan (a \div 2)$. Since the standard American screw thread is 60 degrees, this formula reduces to

$$Q = .577 W. \quad (65)$$

To obtain the *force necessary to shear* or strip a *V* thread from the bolt, multiply the circumference of the bolt at the root of the thread by the length of the nut and by the fibre stress of shearing. To obtain the corresponding force to strip the thread from the nut, multiply the circumference of the outside of the thread by the length of the nut and by the fibre stress of shearing. For an ordinary bolt and nut, the stripping takes place on the threads of the bolt. The length of the thread in action at any time should be such that the force necessary to strip the thread will be greater than the tensional strength of the bolt at the root of the thread.

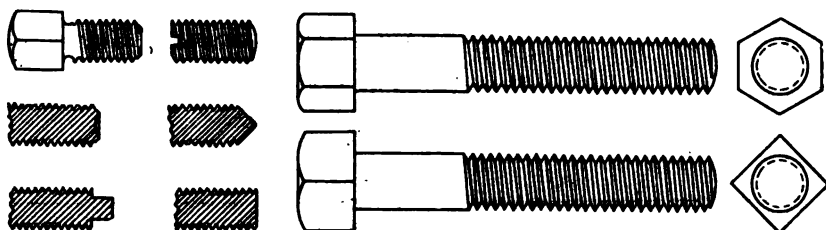
The minimum calculated length for a 1 inch wrought iron bolt in a wrought iron nut is approximately .3 inch. This, in the table, is quoted as 1 inch. The minimum calculated length of thread in the cast iron nut or casing, if used, is approximately .75 inch. This in practice is taken at about $1.5 \times$ diameter of bolt. From this it is seen that the action length of the screw thread is made in practice from two to three times the calculated length.

154. Forms of Standard Threads:—The forms of the standard threads are shown in Fig. 75.



The following tables, No. XVIII, refer to the United States Standard or Sellers Threads.

TABLE XVIII.



SET SCREWS			HEX. HEAD CAP-SCREWS.			SQ. HEAD CAP-SCREWS.		
Short Diam. of Head.	Long Diam. of Head	Lengths (under Head)	Short Diam. of Head.	Long Diam. of Head	Lengths (under Head)	Short Diam. of Head.	Long Diam. of Head	Lengths (under Head)
1/4	.35	3/4 - 3	7/16	.51	3/4 - 3	3/8	.53	3/4 - 3
5/16	.44	3/4 - 3 1/4	1/2	.58	3/4 - 3 1/4	7/16	.62	3/4 - 3 1/4
3/8	.53	3/4 - 3 1/2	9/16	.65	3/4 - 3 1/2	1/2	.71	3/4 - 3 1/2
7/16	.62	3/4 - 3 3/4	5/8	.72	3/4 - 3 3/4	9/16	.80	3/4 - 3 3/4
1/2	.71	3/4 - 4	3/4	.87	3/4 - 4	5/8	.89	3/4 - 4
9/16	.80	3/4 - 4 1/4	13/16	.94	3/4 - 4 1/4	11/16	.98	3/4 - 4 1/4
5/8	.89	3/4 - 4 1/2	7/8	1.01	3/4 - 4 1/2	3/4	1.06	3/4 - 4 1/2
3/4	1.06	3/4 - 4 3/4	1	1.15	3/4 - 4 3/4	7/8	1.24	3/4 - 4 3/4
7/8	1.24	1/4 - 5	1 1/8	1.30	1/2 - 5	1 1/8	1.60	1/2 - 5
1	1.42	1/2 - 5	1 1/4	1.45	3/4 - 5	1 1/4	1.77	3/4 - 5
1 1/8	1.60	3/4 - 5	1 3/8	1.59	2 - 5	1 3/8	1.95	2 - 5
1 1/4	1.77	2 - 5	1 1/2	1.73	2 - 5	1 1/2	2.13	2 1/4 - 5

NOTE.-- The nominal DIAMETER OF SCREW in each of the above is the same as the first column under "Set Screws".

The following tables, No. XIX, referring to the sizes of machine screws, were proposed by a committee of the American Society of Mechanical Engineers, and were acted upon at the May Meeting, 1907.

TABLES XIX MACHINE SCREWS.

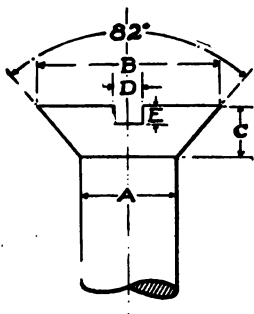


TABLE A, FLAT HEAD SCREWS

A = Diameter of Body

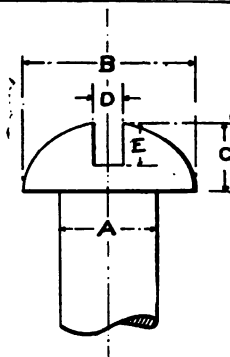
2A — .008 = Diameter of Head

C = $A - \frac{.008}{1.739}$ = Thickness of HeadD = $.173A + .015$ = Width of SlotE = $\frac{1}{3}C$ = Depth of Slot

A					B				
A	B	C	D	E	A	B	C	D	E
.060	.112	.029	.026	.010	.060	.108	.042	.026	.081
.073	.138	.037	.038	.012	.073	.130	.051	.028	.085
.086	.164	.045	.030	.015	.086	.154	.060	.030	.040
.099	.190	.052	.032	.017	.099	.178	.069	.032	.044
.112	.216	.060	.034	.020	.112	.202	.078	.034	.049
.125	.242	.067	.037	.022	.125	.226	.087	.037	.053
.138	.268	.075	.039	.025	.138	.250	.096	.039	.058
.151	.294	.082	.041	.027	.151	.274	.105	.041	.062
.164	.320	.090	.043	.030	.164	.298	.114	.043	.067
.177	.346	.097	.046	.032	.177	.322	.123	.046	.071
.190	.372	.105	.048	.035	.190	.346	.132	.048	.076
.216	.424	.120	.052	.040	.216	.394	.151	.052	.085
.242	.472	.135	.057	.045	.242	.442	.169	.057	.094
.268	.522	.150	.061	.050	.268	.491	.187	.061	.102
.294	.580	.164	.066	.055	.294	.539	.206	.066	.112
.320	.632	.179	.070	.060	.320	.587	.224	.070	.122
.346	.682	.194	.075	.065	.346	.635	.242	.075	.131
.372	.732	.209	.079	.070	.372	.683	.260	.079	.140
.398	.782	.224	.084	.075	.398	.731	.278	.084	.149
.424	.840	.239	.088	.080	.424	.779	.296	.088	.158
.450	.892	.254	.093	.085	.450	.827	.315	.093	.167

TABLE B, ROUND HEAD SCREWS

A = Diameter of Body

B = $1.85A - .005$ = Diameter of HeadC = $.7A$ = Height of HeadD = $1.73A + .015$ = Width of SlotE = $\frac{1}{3}C + .01$ = Depth of Slot

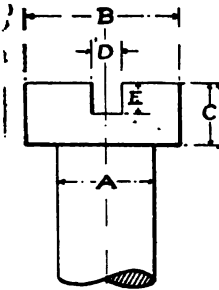


TABLE C, FLAT FILLISTER HEAD SCREWS

- A = Diameter of Body
 B = 1.64A — .009 = Diameter of Head
 C = 0.66A — .002 = Height of Head
 D = 0.173A + .015 = Width of Slot
 E = $\frac{1}{4}$ C = Depth of Slot

C					D					
A	B	C	D	E	A	B	C	D	E	F
.060	.0894	.0876	.085	.019	.060	.0894	.0876	.085	.085	.0496
.078	.1107	.0461	.088	.023	.078	.1107	.0461	.088	.080	.0609
.086	.132	.0548	.090	.027	.086	.132	.0548	.090	.086	.0725
.099	.153	.0633	.093	.032	.099	.153	.0633	.093	.093	.0838
.112	.1747	.0719	.094	.036	.112	.1747	.0719	.094	.094	.0953
.125	.196	.0806	.097	.040	.125	.196	.0806	.097	.098	.1068
.138	.217	.0890	.099	.044	.138	.217	.089	.099	.099	.1180
.151	.2386	.0976	.041	.049	.151	.2386	.0976	.041	.095	.1296
.164	.2599	.1062	.043	.053	.164	.2599	.1062	.043	.071	.1410
.177	.2813	.1148	.046	.057	.177	.2813	.1148	.046	.076	.1524
.190	.3026	.1234	.048	.062	.190	.3026	.1234	.048	.082	.1639
.216	.3459	.1405	.052	.070	.216	.3459	.1405	.052	.086	.1868
.242	.3879	.1577	.057	.079	.242	.3879	.1577	.057	.105	.2097
.268	.4306	.1748	.061	.087	.268	.4306	.1748	.061	.116	.2326
.294	.4731	.1920	.066	.096	.294	.4731	.192	.066	.128	.2554
.320	.5158	.2092	.070	.104	.320	.5158	.2092	.070	.140	.2783
.346	.5584	.2263	.075	.113	.346	.5584	.2263	.075	.150	.3011
.372	.601	.2435	.079	.123	.372	.601	.2435	.079	.162	.3240
.398	.6437	.2606	.084	.130	.398	.6437	.2606	.084	.173	.3469
.424	.6863	.2778	.088	.139	.424	.6863	.2778	.088	.185	.3698
.450	.727	.295	.098	.147	.450	.727	.295	.098	.201	.4024

TABLE D, OVAL FILLISTER HEAD SCREWS

- A = Diameter of Body
 B = 1.64A — .009 = Diam. of Head and
 Rad. for Oval
 C = 0.66A — .002 = Height of Side
 D = 1.73A + .015 = Width of Slot
 E = $\frac{1}{4}$ F = Depth of Slot
 F = .134B + C = Height of Head

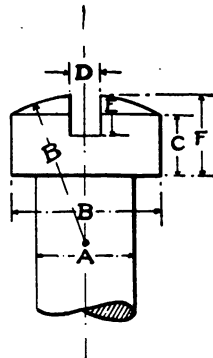
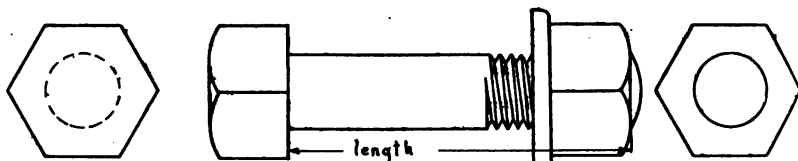


TABLE XX.

Standard Sizes of Machine Bolts and Nuts:



Diam. of screw	Thread per Inch	Diam. of Root of Thrd	Area of Root of Thread	Width of Flat	Short Dia. of Hexagon	Long dia. of Hexagon	Long dia. Square	Thickness of Nuts	Thickness of Heads	Tap Drill
1/4	20	.185	.002658	.0062	1/2	37/64	7/10	1/4	1/4	3/16
5/16	18	.24	.045838	.0074	19/32	11/16	10/12	5/16	19/64	1/4
3/8	16	.294	.067886	.0076	11/16	51/64	63/64	3/8	11/32	5/16
7/16	14	.344	.092940	.0089	25/32	23/32	1 7/64	7/16	25/64	23/64
1/2	13	.4	.125664	.0096	7/8	1	1 15/64	1/2	7/16	13/32
9/16	12	.454	.163984	.0104	31/32	1 1/8	1 23/64	9/16	31/64	15/32
5/8	11	.507	.201886	.0113	1 1/16	1 7/8	1 1/2	5/8	17/32	17/32
3/4	10	.562	.241907	.0125	1 1/4	1 7/13	1 49/64	3/4	5/8	5/8
7/8	9	.621	.283062	.0138	1 7/16	1 21/32	2 1/32	7/8	23/32	3/4
1	8	.687	.326226	.0156	1 5/8	1 7/8	2 19/64	1	15/16	27/32
1 1/8	7	.74	.373578	.0173	1 13/16	2 3/32	2 9/16	1 1/8	29/32	31/32
1 1/4	7	1.065	.49082	.0173	2	2 5/16	2 53/64	1 1/4	1	1 3/32
1 3/8	6	1.16	1.05629	.0208	2 3/16	2 17/32	3 3/32	1 3/8	1 3/32	1 3/16
1 1/2	6	1.284	1.29485	.0208	2 3/8	2 3/4	3 23/64	1 1/2	1 3/16	1 9/32
1 5/8	5 1/2	1.389	1.51283	.0227	2 9/16	2 31/32	3 5/8	1 5/8	1 9/32	1 13/32
1 3/4	5	1.491	1.74647	.028	2 3/4	3 3/16	3 57/64	1 3/4	1 3/8	1 1/2
1 7/8	5	1.616	2.05107	.026	2 15/16	3 13/32	4 5/32	1 7/8	1 13/32	1 5/8
2	4 1/2	1.712	2.30205	.0277	3 1/8	3 5/8	4 27/64	2	1 9/16	1 3/4
2 1/4	4 1/2	1.962	3.05424	.0277	3 1/2	4 1/16	4 61/64	2 1/4	1 3/4	1 31/32
2 1/2	4	2.176	3.71385	.0312	3 7/8	4 1/2	5 21/64	2 1/2	1 15/16	2 3/16
2 3/4	4	2.426	4.61845	.0312	4 1/4	4 29/32	6	2 3/4	2 1/8	2 7/16
3	3 1/2	2.629	5.48840	.0357	4 5/8	5 3/8	6 17/32	3	2 5/16	2 5/8
3 1/4	3 1/2	2.879	6.49890	.0357	5	5 13/16	7 1/16	3 1/4	2 1/2	2 29/32
3 1/2	3 1/4	3.1	7.54768	.0384	5 3/8	6 7/64	7 39/64	3 1/2	2 11/16	3 1/32
3 3/4	3	3.317	8.64135	.0418	5 3/4	6 21/32	8 1/8	3 3/4	2 7/8	3 11/32
4	3	3.567	9.99302	.0413	6 1/8	7 3/32	8 41/64	4	3 1/16	3 19/32
4 1/2	2 7/8	3.798	11.3292	.0436	6 1/2	7 9/16	9 3/16	4 1/4	3 1/4	3 13/16
4 1/2	2 3/4	4.028	12.8693	.0436	6 7/8	7 31/32	9 3/4	4 1/2	3 7/16	4 1/32
4 3/4	2 5/8	4.266	14.2268	.0476	7 1/4	8 13/32	10 1/4	4 3/4	3 5/8	4 9/32
5	2 1/2	4.48	15.7638	.05	7 5/8	8 27/32	10 49/64	5	3 13/16	4 1/2
5 1/2	2 1/2	4.73	17.5716	.05	8	9 9/32	11 23/64	5 1/4	4	4 3/4
5 1/2	2 3/8	4.953	18.2284	.0526	8 3/8	9 23/32	11 7/8	5 1/8	4 3/16	4 31/32
5 3/4	2 3/8	5.203	21.2535	.0526	8 3/4	10 5/32	12 3/8	5 3/4	4 3/8	5 7/32
6	2 1/4	5.423	23.0892	.0555	9 1/8	10 19/32	12 15/32	6	4 9/16	5 7/16

155. Standard Machine Bolts and Nuts:—

Hexagon heads are generally finished.

Square heads are generally not finished.

A washer is frequently placed between the nut and the clamped piece.

Where a lock nut is used the thicker nut is placed on the outside.

The diameter of the screw at the root of the thread is 0.8 of the nominal diameter of the screw for small screws, and 0.9 of the nominal diameter for large screws.

Pitch, $p = 0.24 \sqrt{D} + 0.625 - 0.175$, where D is the outside diameter of the screw.

156. Stud Bolts:—If two castings are to be bolted together, and one be tapped for a screw fastening, a cap screw may be used where the pieces will seldom need separation. If, however, the pieces frequently need to be taken apart, the stud bolt should be used. The brittleness of the cast iron thread renders it more susceptible to wear, hence the above statement.

A stud bolt, (See Fig. 79, C and D) consists of a rod threaded at both ends. One end is screwed into a tapped hole in one of the pieces to be held together, and the other receives a nut which presses against the clamped piece.

For a cap screw or stud bolt the depth of the threaded hole depends largely upon the material; in all cases, its depth should be as great as the diameter of the screw. When the material tapped into is weaker than that of the screw, as when a steel screw is screwed into cast iron, the threaded depth of the hole should be made $1\frac{1}{2}$ to 2 times the diameter of the screw. Under usual conditions a stud does not need as long a thread in the tapped hole as a cap-screw serving the same purpose.

157. Eye-Bolts:—In designing eye-bolts the combined cross sectional area at the sides of the eye should be made equal to about 1.5 times the least sectional area of the threaded portion of bolt, to allow for uneven distribution of stress. For application see Air Hoist, Design No. 3, Par. 214.

158. References for Screw Gearing:—Reuleaux, "The Constructor," pages 139-144, 50-60; Jones, "Machine Design," Part II, pages 150-200; Low and Bevis, "Machine Drawing and Design," pages 86-100; Benjamin, "Machine Design," pages 54-58.

159. Problems.—

(1). A worm and wheel are used in connection with a rack and pinion to raise the table of a drill press. The worm wheel and pinion are keyed to the same mild steel shaft. The handle, keyed to the worm shaft, is 10 inches long, the worm is single threaded, and the pitch radius of the worm wheel is approximately 5 inches. The

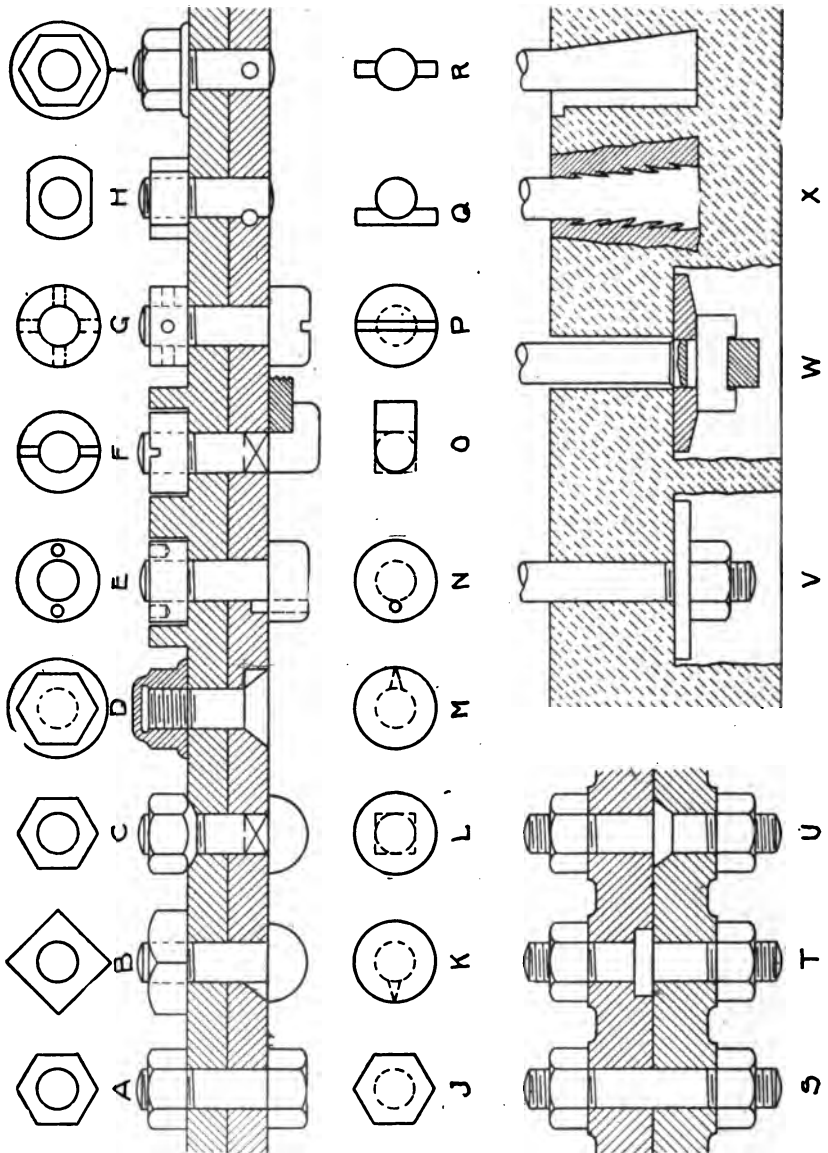
160. Bolt Fastenings:—

FIG. 76.

table, etc., weighs 600 pounds. What weight could be lifted on the table by a turning force of 15 pounds, applied to the handle, if 60 per cent of the work applied is lost in friction? Determine the pitch

161. Nut Fastenings:—

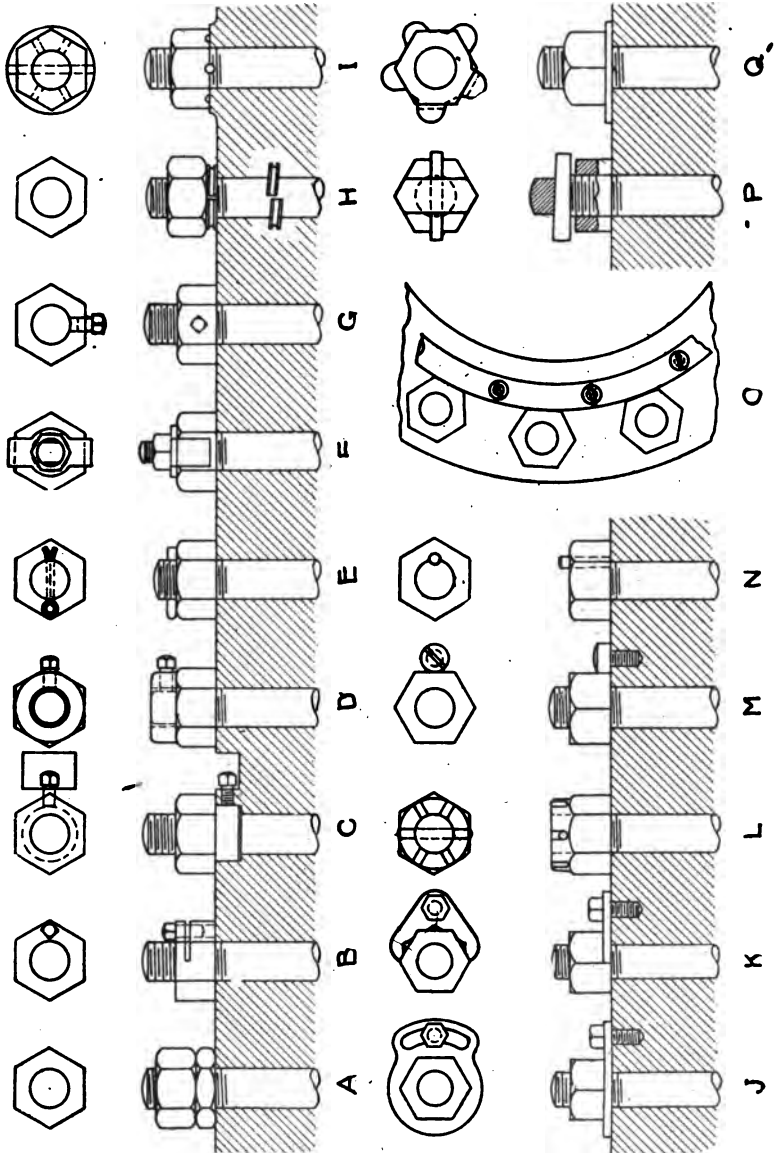


FIG. 77.

of the worm wheel, the pitch of the pinion, and the diameter of the shaft connecting them.

(2) In a screw press such as is shown in A, Fig. 74, the square-

threaded screw is fed downward, 1 inch in $2\frac{1}{4}$ revolutions, by a worm-wheel, acting as a nut, *D*, Fig. 74. This worm-wheel has 76 teeth and is driven by a single-threaded worm. The hand wheel on the worm shaft is 18 inches in diameter. The platen on the lower end of the screw measures 10×12 inches and exerts a pressure of 130 pounds per sq. in. The coefficient of friction for the screw and worm-wheel is 0.16; and the coefficient of friction for the worm-wheel and frame is 0.19, with a pressure not to exceed 1000 pounds per sq. in.

Determine the force which must be applied at the rim of the hand wheel in order to lower the platen against the above pressure, if 65 per cent of the force applied is lost in the friction of the worm and wheel, in addition to the friction loss between the worm-wheel and frame and worm-wheel and screw. Find the diameters of screw and worm shaft if they are made of wrought iron. Determine the pitch, length, external diameter and diameter at root of thread for the worm. What will be the pitch diameter; width of face; and length and outside diameter of hub for worm-wheel?

(3) A worm wheel having 48 teeth forms the nut of a screw of $\frac{3}{4}$ inch pitch, single thread, (*D*, Fig. 74); a single threaded worm driven by a lever 16 inches long gears with the worm wheel. The entire mechanism is assumed to be 75 per cent efficient.

(a) Find the pressure exerted by the end of the screw, and the pitch of the worm wheel when 20 pounds is applied to the lever.

(b) If the screw was $\frac{3}{4}$ inch pitch, double thread, what would the results be?

(4) In an old English planer the table carrying the work has a travel of 10 feet, and is driven by a mild steel screw. A spur gear approximately 6 inches in diameter, is keyed to one end of the screw. This spur wheel is in turn driven by a 13 inch (approximate) pinion, keyed on the same shaft with a driving pulley 26 inches in diameter with a $5\frac{1}{2}$ inch face, carrying a double leather shop belt, at 200 revolutions per minute. The resistance offered by the tool in making cut is 1600 pounds. The table weighs 4500 pounds, and the casting being planed 1800 pounds. The coefficient of sliding friction between the table and its guides is assumed to be 0.18, while the coefficient of friction for the screw and thrust bearing is 0.14; the allowable pressure for the thrust bearing being 800 pounds per square inch. Maximum distance between thrust bearing and nut actuating table, 11 feet. The small diameter of the thrust bearing is made equal to the diameter of the screw at root of thread. Assuming the spur gears to have an efficiency of 85 per cent, and the width of belt to be $\frac{1}{2}$ inch less than pulley face; Find:—(a) Diameters for thrust bearings, pitch of screw and length of nut on screw, also pitch of spur gears; (b) External diameter of screw, exact pitch diameters for spur gears, and diameter of wrought iron shaft for belt pulley; (c) The maximum economical cutting speed for tool.

CHAPTER IV.

Cylinders, Cylinder Heads and Fastenings.

162. Cylinders with Thin Walls Subjected to Internal Pressures.—Assume a theoretically perfect cylinder, with the following notation: radius r , diameter d , length l , and thickness t . If this cylinder be subjected to an internal pressure of p pounds per square inch, it will have a tendency to cause a rupture of the shell either parallel with or transversely across its axis. In the first case

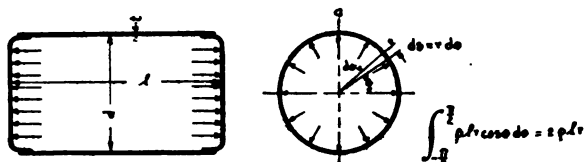


FIG. 78.

the bursting force tending to split the cylinder along the line $a-b$ is $2 p l r$, Fig. 78, also Church, par. 169; and the resistance to this force is the strength of the two sides, equal to $2 t l f$. Forming this into an equation it gives,

$$2 t l f = 2 p l r, \text{ or } t = p d \div 2 f. \quad (1)$$

The force tending to cause the cylinder to fail transversely is the pressure on the head, equal to $\pi r^2 p$, and the resistance to this is the strength of the shell, equal to $2 \pi r t f$ (approx.),

hence: $2 \pi r t f = \pi r^2 p, \text{ or } t = p d \div 4 f \text{ (approx.)} \quad (2)$

This shows that the cylinder is twice as strong to resist transverse rupture as it is to resist longitudinal rupture. Formula (2) is satisfactory for all cylinders where the shell is very thin compared to its diameter, as found in boilers. It is understood that the radius of the shell is here taken as a mean radius only. For cylinders with thick shells the resisting factor would become $\pi (D^2 - d^2) \div 4$ where D and d are the diameters, respectively, of the outside and the inside of the shell. Since the tendency of any cylinder, then, would be to first give way along its length, formula (1) is generally used.

163. Boiler Shell:—In applying wrought iron or steel to form cylindrical shells it is necessary to unite the ends of the plates by welding or riveting; in either case reducing the ultimate strength of the shell. Welding is resorted to in tubes and pipes, giving efficiencies varying between 50 and 90 per cent of the solid plate of the same thickness. The average efficiency may be taken at 70 per

cent. Riveted joints are used on shells of large diameter such as boiler shells. The efficiencies of the joints vary, according to the Hartford Inspections and Insurance Co., between 70 per cent, for a double riveted lap joint and 86 per cent for a triple riveted butt joint having the same sized rivets. When the kind of joint is not stated, the efficiency may be taken at 70 per cent of the solid plate. For details of all forms of riveted joints for boilers see Ryerson's catalog of Boiler Diagrams; Jones' Machine Design, Part II, Chap. 12. Low and Bevis, Chap. 21; Unwin, Part I, Chap. 4; The Constructor, Sec. III. Chap. 1.

The *thickness* of a boiler shell should be worked out from formula (1) by inserting the factor c as the efficiency of the joint.

Application:—What should be the thickness of a 60 inch boiler shell if the steam pressure is 150 pounds gauge and the material of the shell is open hearth steel having an ultimate tensile strength of 60,000 pounds per square inch? With a factor of safety of 6, and 70 per cent. efficiency of the joint, formula (1) gives

$$t = \frac{p d}{2 f c} = \frac{150 \times 60}{2 \times 10000 \times .70}; t = .643 \text{ inches.}$$

A plate of 11-16 inches will be necessary to fulfill these conditions.

164. Machine or Cast Cylinders:—Cast iron is the material used in nearly all cases where cylinders require an inside finish for pistons or moving parts. Where the pressures are high and the thickness of the cast iron walls would be too great, steel castings are used instead of the gray iron castings. All castings are subject to hidden flaws and defects, and a great deal of uncertainty accompanies their use. Because of this uncertainty and because cylinders require re boring occasionally, designers estimate the thickness of cast cylinders by formula (1) allowing very large factors of safety, or as is the case in most *steam* and *air cylinders*, they use empirical formulas. One of the most satisfactory of these formulas for cast iron cylinders is quoted in the Trans. A. S. M. E. Vol. 18, page 741.

$$t = .05 D + .3'' \quad (3)$$

where D is the diameter of the cylinder in inches. This formula should be applied to cylinders using not over 125 pounds gauge pressure.

165. Cylinders with Thick Walls Subjected to Internal Pressures:—If the walls of a cylinder are thick compared to the cylinder diameter, the stresses in the metal, from internal pressure, will vary between the interior and exterior surfaces, being greatest on the interior fibres. The following formulas are used and give approximately the same result when designing cylinders between 6 and 18 inches diameter, and carrying pressures between 300 and 1000 lbs. per square inch with a fibre stress of 2000 pounds per square

inch. As the ratio of f to p becomes smaller and approaches 1 these formulas give absurd results. It is well to keep this ratio in all cases as great as 2.

$$\text{Barlow,} \quad t = \frac{pd}{2(f-p)} \quad (4)$$

$$\text{Grashof,} \quad t = \frac{d}{2} \left(\sqrt{\frac{3f+2p}{3f-4p}} - 1 \right) \quad (5)$$

In which t = thickness of walls in inches; p = gauge pressure in pounds per square inch; d = internal diameter in inches and f = allowable fibre stress in pounds per square inch.

Another very satisfactory formula is given by

$$\text{Perry,} \quad f = p \left(\frac{r_1^2 + r^2}{r_1^2 - r^2} \right)$$

Where r_1 = external radius; r = internal radius; and f and p as above.

Where a cylinder is so designed that it may be ground instead of bored it is possible to chill the inner surface to such an extent that the particles are in a state of compression and hence will yield more to the applied pressure and cause a more uniform stress throughout the shell when in service. This would make the condition of the design very similar to that of a thin cylinder. For discussions on thick cylinders see Kent, page 287; Unwin, Part 1, par. 26 a; Perry, par. 275; Cotterill, par. 214.

166. Cylinders with Thin Walls Subjected to External Pressures:—An empirical formula developed by Fairbairn for the collapsing pressure of thin wrought iron tubes is discussed in Cotterill, par. 181; Unwin, Part 1, par. 41, and Kent, page 264. Let p = differential pressure in pounds per square inch, t = thickness of shell in inches, d = diameter in inches and l = length in inches, then

$$p = 9672000 \, t^2 \div ld \quad (6)$$

This should not be used for extreme cases nor for thickness less than $\frac{3}{8}$ of an inch.

Lloyds Register contains the following formula,

$$p = 1075200 \, t^2 \div ld \quad (7)$$

It will be seen that this is the same as (6) with a factor of safety of 9.

167. A Thin Sphere Subjected to Internal Pressure would have a tendency to rupture along the circumference of its diameter. The force exerted would be $p \pi d^2 \div 4$ and the resistance $\pi d t f$ approximately, hence we have with the same notation as above,

$$t = pd \div 4f \quad (8)$$

It will be seen that (8) is the same as (2) and makes the sphere twice as strong as the cylinder of the same diameter and thickness

of shell; hence a spherical boiler head would be as strong as the shell if it was one-half its thickness. When this formula is applied to wrought iron or steel work it will be necessary to account for the efficiency of the riveted joint.

Dished heads, for boiler work, have the same strength to resist rupture as the boiler shell of the same thickness, if the radius of curvature is equal to the diameter of the boiler shell. This may be obtained by inspection from formulas (1) and (8). The *camber* of the head will be $h = .134 d$; where d = diameter of shell and h = height of camber, both given in same units. Low and Bevis, page 305.

168. Strength of Flat Plates:—The relation existing between the thickness of a flat plate and its fibre stress and deflection when subjected to known conditions of loading can be obtained from the formulas given in Table XXI. See also, *American Machinist*, 1906, page 683.

Of all the conditions here shown the second one is the one most often found in practice. This formula would cover most forms of cylinder heads. Kent discusses the subject on pages 284 and 794 of his *Pocket Book* and suggests a number of empirical rules to be used instead of the rational ones, which he says, in many cases give too large results.

Benjamin gives the following empirical formulas as the result of experiments made at the Case School of Applied Science on rectangular cast iron plates with *load concentrated at the centers*

$$\text{Plate supported at edges. } W = 272 \frac{S t^2}{l^2 + b^2} \quad (9)$$

$$\text{With edges fixed. } W = 442 \frac{S t^2}{l^2 + b^2} \quad (10)$$

Where W = breaking load, in pounds;

S = modulus of rupture, pounds per square inch = 33000;

t = thickness of plate, in inches;

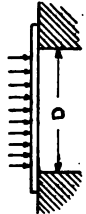
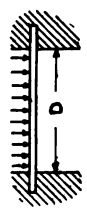

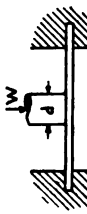
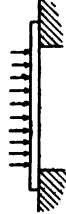

l = length of plate, in inches;

b = breadth of plate, in inches.

A round plate may be considered as square, l = diameter, without sensible error.

Where approximate values are considered satisfactory, it is good practice to take the thickness of the head at the edge 25 per cent greater than the thickness of the cast iron cylinder walls. The central part of the head may be made thinner than the edge, and is sometimes set off from the plane of the outer flange. The thickness of this central portion should be about the thickness of the cylinder walls.

TABLE XXI.

	CONDITION	AUTHOR	FIBRE STRESS	DEFLECTION	NOTES
	Flat circular plate. Uniform load. Supported at edge.	GRASHOF	$f = \frac{3}{16} \frac{D^2 w}{t^3}$	$\Delta = \frac{1}{24} \frac{D^4 w}{t^3 E}$	t = Thickness of plate w = Load in "per sq. in."
		MERRIMAN	$f = \frac{3}{16} \frac{D^2 w}{t^3}$ $C = 1$		
	Flat circular plate. Uniform load. Fixed at edge.	GRASHOF	$f = \frac{1}{6} \frac{D^2 w}{t^3}$	$\Delta = \frac{1}{96} \frac{D^4 w}{t^3 E}$	
		MERRIMAN	$f = \frac{8}{32} \frac{D^2 w}{t^3}$ $C = 1$		
	Flat circular plate. Concentrated load. Within circumference πd . Supported at edge.	GRASHOF	$f = \left[\frac{4}{3} \log \left(\frac{D}{d} \right) + 1 \right] \frac{W}{\pi t^3}$ $f = C \frac{W}{\pi t^3}$	$\Delta = .13 \frac{D^4 W}{t^3 E}$	W = Total load $D/d = \left[\frac{10}{10} \right] \left[\frac{30}{30} \right] \left[\frac{40}{40} \right] \left[\frac{50}{50} \right]$ $C = \left[\frac{4.27}{4.27} \right] \left[\frac{5.0}{5.0} \right] \left[\frac{5.53}{5.53} \right] \left[\frac{5.82}{5.82} \right]$ w = Uniform pressure in "per sq. in." area d^2 . For $C, 1, d = \frac{1}{2}$. For $w, 1, \pi d$ Steel $a = \frac{1}{2}$.
		MERRIMAN	$f = \frac{2 \log (1-d)}{t^3} \left(\frac{Dd}{2} - \frac{d^2}{4} \right)$		
	Flat circular plate. Concentrated load. Within circumference πd . Fixed at edge.	GRASHOF	$f = \left[\frac{4}{3} \log \left(\frac{D}{d} \right) \right] \frac{W}{\pi t^3}$ $f = C \frac{W}{\pi t^3}$	$\Delta = .082 \frac{D^4 W}{t^3 E}$	W = Total load $D/d = \left[\frac{10}{10} \right] \left[\frac{30}{30} \right] \left[\frac{40}{40} \right] \left[\frac{50}{50} \right]$ $C = \left[\frac{3.27}{3.27} \right] \left[\frac{4.0}{4.0} \right] \left[\frac{4.53}{4.53} \right] \left[\frac{4.82}{4.82} \right]$ w = Uniform pressure in "per sq. in." area d^2 . For $C, 1, d = \frac{1}{2}$. For $w, 1, \pi d$ Steel $a = \frac{1}{2}$.
		MERRIMAN	$f = \frac{2 \log (1-d)}{2 t^3} \left(\frac{Dd}{2} - \frac{d^2}{4} \right)$		
	Elliptical flat plate. Uniform load. Supported at edge. Semi-axes A and B .	BACH MERRIMAN	$f = q \frac{A^2 B^2 w}{(A^2 + B^2)^2 t^3}$		For $C, 1, a = \frac{1}{2}$. For $w, 1, \pi d$ Steel $a = \frac{1}{2}$. For fixed plates $1-a$ for $C, 1 = \frac{1}{2}$; For $w, 1, \pi d = a, b$.
		BACH MERRIMAN	$f = q \frac{A^2 B^2 w}{(L^2 + M^2)^2 t^3}$		
	Rectangular flat plate. Uniform load. Supported or fixed. Sides L and M .	BACH MERRIMAN			$a = \frac{1}{2}$ for edges supported. $a = \frac{1}{2}$ for edges fixed.

Where *extra heavy* pressures are involved, it is not safe to depend upon the approximate rules, but each case should be worked out by an approved formula, and then checked up by the best current practice.

169. Method of Fastening Between Cylinder and Cylinder Head: In most cylinder work it is necessary to cap the ends. The form of this capping differs according to the materials employed and the use to which the cylinder is to be applied. The wrought iron or steel cylinder, for example, is used mostly on boilers and is capped with a head of like material. This head is flanged and riveted to the shell. If the head is dished, as shown in Fig. 79, A, to the radius of the shell diameter, the thickness of the material may be the same as that of the shell and need not be stayed, but if the head is flat as in B, it is made somewhat thicker than the shell and in addition is stayed. The staying may be done by *through stays* or by *angle stays* depending upon the interior construction and requirements of the boiler. For more details on staying see notes on "Engine and Boiler Design," or Peabody & Miller, "Steam Boilers."

Cast Cylinders for steam engines, compressors, hoists, etc., would have a fastening similar to C, the heads being bolted to flanges that are an integral part with the cylinder body. These fastenings may be by ordinary bolts, stud bolts, or cap screws. For rough work the bolt is preferred. For finished work the stud bolt is preferred. The cap screw would be used only in special cases where the cylinder head would seldom be removed, since frequent removals would tend to ruin the threads in the cast iron flange.

Concerning the *number* and the *size of the bolts*; it is better to have a number of bolts of small cross sectional area, than a less number of bolts of large area, for if one bolt proves defective the extra stress is more readily absorbed by those near to it. The number of bolts used will approximate closely to the nominal diameter of the cylinder in inches, i. e. a 6 inch cylinder, 6 bolts; a 24 inch cylinder, 24 bolts, etc.

In cylinders carrying a pressure of 75 to 100 pounds gauge, it is not advisable to use less than a $\frac{5}{8}$ inch bolt because of the liability of injury when setting the nut. All bolts should be figured at the root of the thread. All bearing surfaces should be accurately finished. The joint should be tight under pressure, either by being ground to a fit or by facing with some soft thin packing.

A modification of C is shown in F where eccentric head bolts are used instead of the ordinary stud bolt.

To cap a pipe cylinder with a cast iron head it is necessary to first fasten a flange to the pipe by screwing or expanding the pipe into it, and then fasten the head to this flange as shown in D, G and H. In D the pipe should be screwed through the flange and faced off with it. G and H should be faced off in like manner and in ad-

dition should have soft metal packing swaged into dovetail grooves as at *s*.

A *Brass or Copper tube cylinder* may be capped as shown at *E*. This metal being too thin to be threaded it is faced off square at each end and bolted between cast heads. A soft copper or lead packing may be applied to the bottom of each finished groove to cut off leakage. Such cylinders are very common in small air hoists.

170. References for Cylinders, Cylinder Heads, and Fastenings:—Benjamin, "Machine Design," pages 25-53; Low and Bevis, "Machine Drawing and Design," pages 328-330, 309-318; Peabody and Miller, "Steam Boilers," pages 182-186, 148-163.

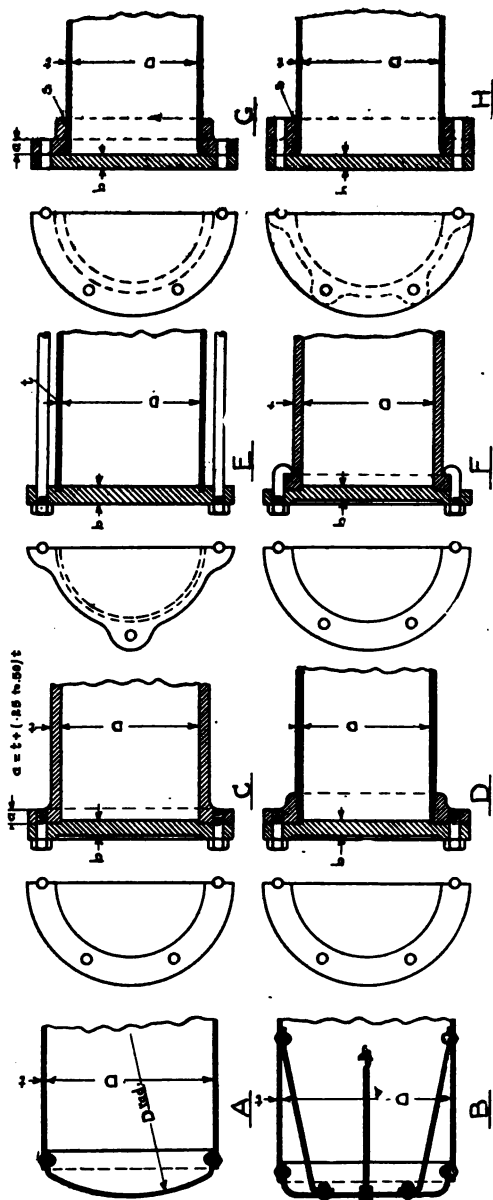
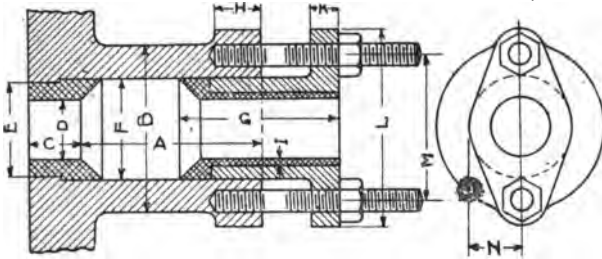


FIG. 79.

171. Standard Sizes of Stuffing Boxes:—



D = DIAM. OF ROD

H = .30 D + .5"

A = 1.60 D + 1.5"

I = .04 D + .1875"

B = 1.75 D + 1.125"

K = .25 D + .25"

C = .10 D + .75"

L = 2.25 D + 1.75"

E = 1.25 D + .375"

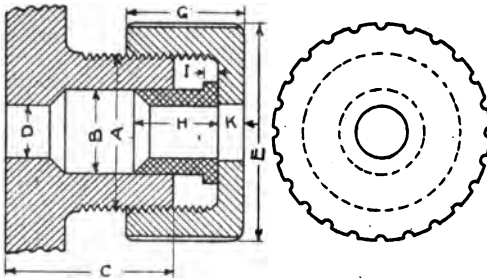
M = 1.60 D + 1.25"

F = 1.25 D + .625"

N = .75 D + .375"

G = 1.50 D + 1."

USE 2 BOLTS IN GLAND ON RODS UP TO 3.5"
DIAM. ABOVE THAT SIZE MAKE GLAND
ROUND AND PUT IN 3 BOLTS.



D = DIAM. OF ROD

I = .25 D + .0625"

A = 2.5 D + .5"

K = .50

B = 1.5 D + .125"

C = .30 D + .25"

E = 3.5 D + .625"

G = 2 D + .25"

H = 1.5 D + .25"

THIS STYLE USED ON RODS UP TO 1 1/4"
DIAM. THREADS PER INCH SAME
AS FOR BOLT OF DIAM. OF ROD.

FIG. 80.

172. Suggestions on the Shapes of Pistons:—

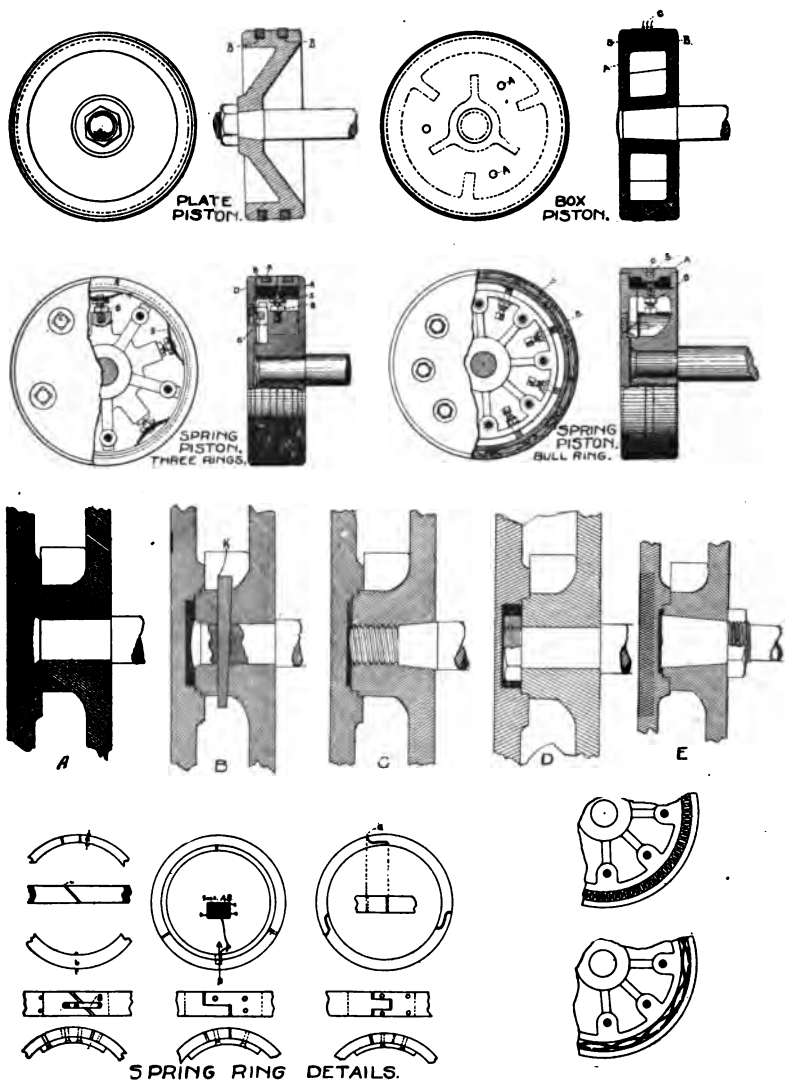


FIG. 81.

CHAPTER V.

General Suggestions on the Designs.

173. Drawings:—The following dimensions are given as the cutting sizes of the sheets. The designer is at liberty to make his own selection from these sizes. It is suggested however, that the sheets be taken as small as will admit of a clear and distinct set of drawings.

24" X 36"	—Size A
18" X 24"	— " B
12" X 18"	— " C
9" X 12"	— " D

Scale:—Any scale may be taken which will show clearly all the details and give a good arrangement on the sheet. Details may have different scales on the same sheet if so desired. When this is done each detail should have the scale given.

Border Line:—A margin of one-quarter of an inch should be left between the border line and the edge of the finished sheet on the top, bottom and right end, and one-half inch on the left end to allow for punching and fastening.

Name Plate:—Make the name plate or title at the lower right hand corner to cover a space $2\frac{1}{4}$ inches X $3\frac{1}{2}$ inches. No border line need be drawn around this name plate. The name plate should be considered standard and should be worked up after the plan of those shown on the sample plates. It would be well for each designer to make a standard corner plate to be used below the various tracings when working up this part.

All drawings will be carefully worked up in pencil and turned in to the instructor. The instructor will then give them to another designer who will be responsible for the checking. This checking will be done in the form of notes on a separate paper and attached to the drawings. These are then returned to the designer for approval and corrections. The designer then traces his drawings, or such part of them as may be selected by the instructor, and after obtaining the signature of the checker to them submits the same with the checkers' notes to the instructor for approval.

Each designer should have experience not only in planning and executing well his own designs, but he should take up designs of other men and offer suggestions and criticisms upon their work. One way to obtain this experience has been suggested above.

In checking up the work of another man the following points should be observed:

(1). General appearance of the design relative to workmanship and execution, arrangement of drawings, notes, dimensions, etc.

(2). General design relative to proportion, and strength and arrangement of parts. This is to be merely the checker's impression and need not require the checking of the original calculations.

No drawing should be retained longer than one exercise and at the completion of the checking should be returned to the designer. It is estimated that any set of drawings may be checked in this way within two hours time. No notes or marks will be made on the drawings but special paper will be provided for this purpose. In looking over the drawings finally, the instructor will give credit to the work of the checker as well as to that of the designer.

In all this work Jamison's *Elements of Mechanical Drawing* will be used as reference concerning arrangement of views, sectioning, cross hatching, lettering, and the like.

Every dimension should be clearly shown so that no measurements need be taken by scale from the drawing.

All dimensions should be given in round vertical figures, heavy enough to print well. No diagonal-barred fractions, thin or doubtful figures will be accepted.

All dimensions below 3' — 0" should be given in inches.

All dimension lines should be made as light as will insure good printing and should have a central space for figures.

All dimensions should read in the direction of the arrows.

Avoid crowding the dimensions to the center of any detail. A much better way is by the use of projected lines as shown by Fig. 82.

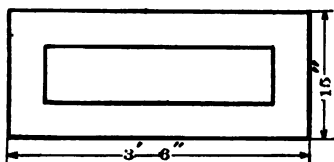


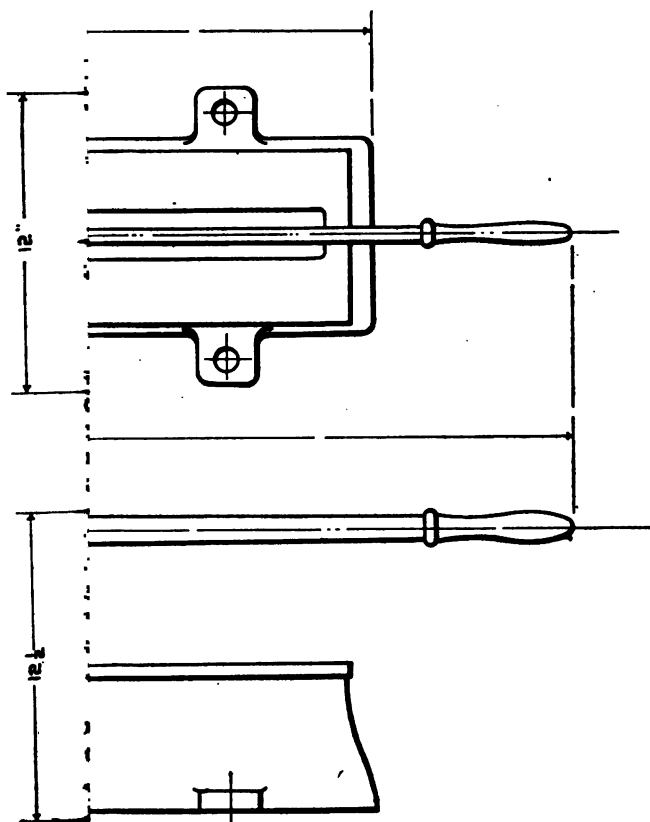
FIG. 82.

All detailed pieces should be accompanied by a *shop note* or *call* as "C. I. One wanted;" "M. S. Two wanted"; "Finish all over," "Turned for a shrinking fit," etc., etc.

The following abbreviations will be considered satisfactory in these calls:—

C. S. Cast Steel	f. Finish. (see sheets of details.)
C. I. Cast Iron.	B. b. t. Babbitt Metal.
W. I. Wrought Iron.	D. Diameter.
M. S. Machine Steel.	R. Radius.

174. Calculations:—Each designer is expected to draw up a report in parallel with the design. This report will contain such free-hand sketches as relate to the calculations, also a full report of the calculated sizes and accepted sizes of the different parts of the design, and will be submitted in a manilla cover with the finished tracings and drawings.

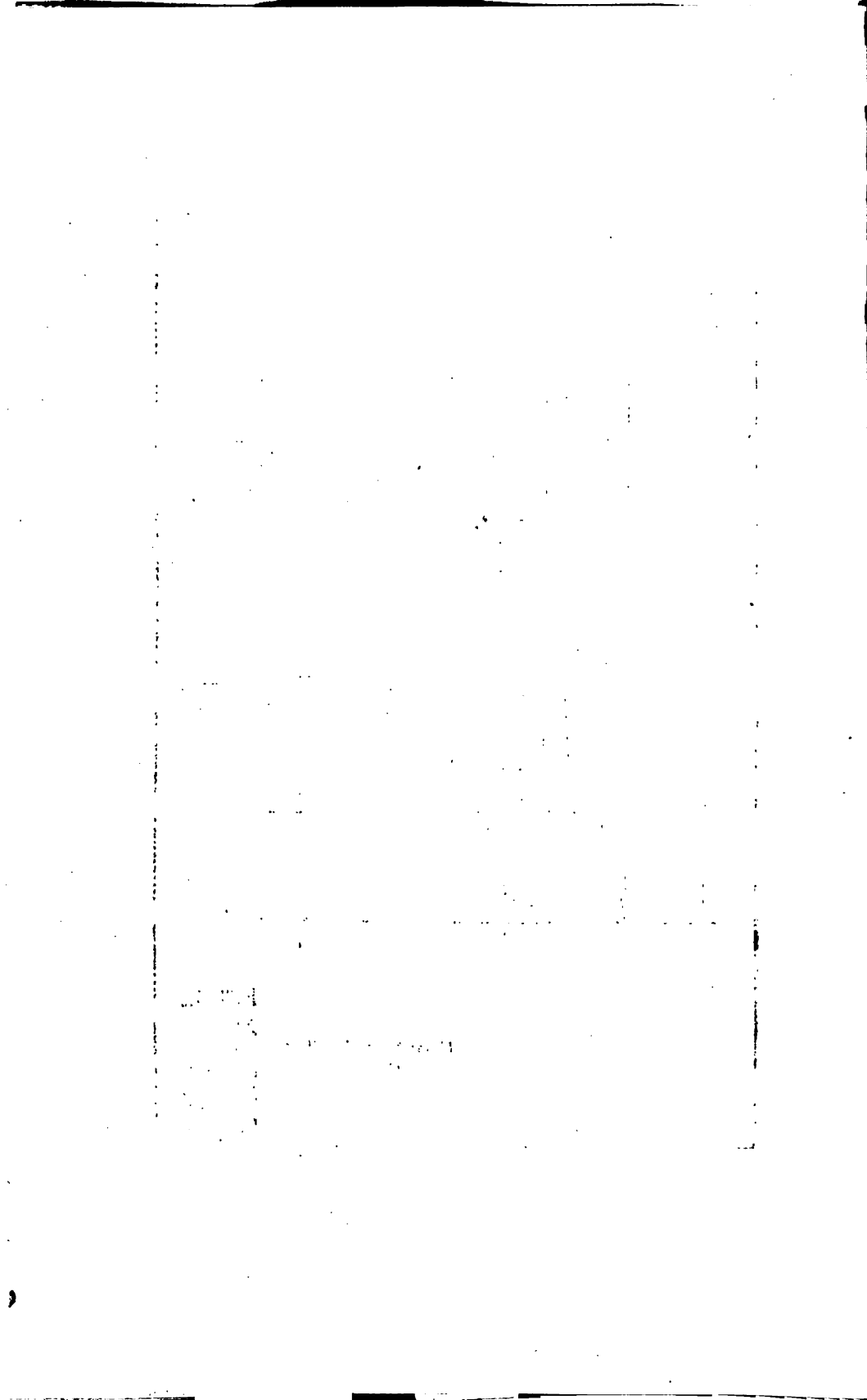


TOGGLE JOINT PRESS

ASSEMBLY

Scale - 1/4 size.

Purdue University, Lafayette, Ind.
 Drawn by E. L. Smith.
 Checked by J. Riley.
 Approved J. H. H. C-191.
 425-03.



DESIGN NO. 1.

175. Introductory Statement:—In entering the subject of Machine Design it has been found that much time is saved by assuming some simple machine and illustrating methods in design by a fairly complete analysis of all the important theoretical calculations. Such a lay-out at once gives the scope of the work and protects the beginner from so much "working in the dark." An assignment is then made, differing in a lesser or greater degree from the illustrated design and a complete analysis required of all parts of the machine. After the student's experience with the first design he will need the second one developed less elaborately and possibly the third one not at all.

Design No. 1 is meant especially to cover static forces; or, simple applications of members in tension, compression, flexure and shear. A good illustration of this is the toggle joint press. Machines of this class are sometimes used in forming thin sheets of copper and brass into articles for ornamental purposes, consequently it is a useful tool. Plates C-191, C-192, and C-193 show a design of a small machine and are inserted to give an idea as to the arrangement of the drawings. The design was worked up on three 12 inch \times 18 inch sheets; two of details and one assembly view.

It is urged that the designer regard these sheets merely as illustrative of a good drawing room job and that, from the standpoint of ideas he will cultivate originality and make a design as nearly independent as possible.

Alternative Designs will be found at the end of this chapter. These may be substituted for the regular designs if preferred.

176. Analysis of the Forces Involved:—Referring to Plate C-191, it will be seen that the acting forces can be represented in the following force diagram, with the direction of the forces represented by arrows.

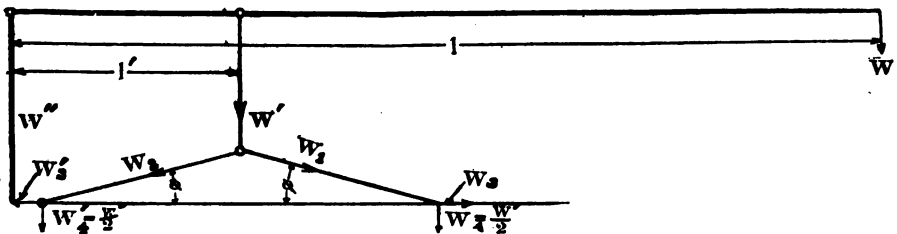


FIG. 83.

Each designer will be given a value for W , l , l' and ϕ . In all the designs ϕ may be taken at 10° , assuming that the maximum load will be carried at this position and that the lever arm will then be horizontal.

In the assignments for a number of designs the range of values will be as follows:—

$W = 200, 300, 400 \dots 1000$ pounds
 $l = 4, 4.5, 5, 5.5 \dots 10$ feet.
 $l' =$ for large sizes, 6, 8, 10. 12 inches.
 for small sizes 6, 6.5, 7. 8 inches.

Selecting for our analysis the following values: $W = 100$ pounds; $l = 5$ feet 3 inches; $l' = 7$ inches; and $\phi = 10$ degrees, we have from the force diagram

$$W' = \frac{W l}{l'} = \frac{100 \times 63}{7} = 900 \text{ pounds}$$

$$W'' = \frac{W (l - l')}{l'} = \frac{100 \times 56}{7} = 800 \text{ pounds.}$$

$$W_1 = W_2 = \frac{W'}{2 \sin. \phi} = \frac{900}{.3473} = 2591.4 \text{ pounds}$$

$$W_3 = W_1 \cos \phi = 2591.4 \times .98481 = 2552 \text{ pounds.}$$

$$W_4 = \frac{900}{2} = 450 \text{ pounds.}$$

177. Lever:—This will be designed as a beam in flexure, Par. 48. The designer must here decide if the beam is to have parallel

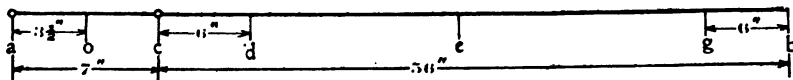


FIG. 84.

sides, in which case b in the modulus for the rectangular section would be constant for all sections, or taper sides, in which case a certain ratio of b to h would be used. The best way to decide which to use is to find the size of the sections at g and c for each case. Assuming $f = 8000$ for wrought iron, $b = 1$ and disregarding the hole at c which has little effect, see par. 179, our formula $W = f Z$ becomes

$$(\text{Section at } g) 100 \times 6 = 8,000 h^2 \div 6; h = .67 \text{ inch.}$$

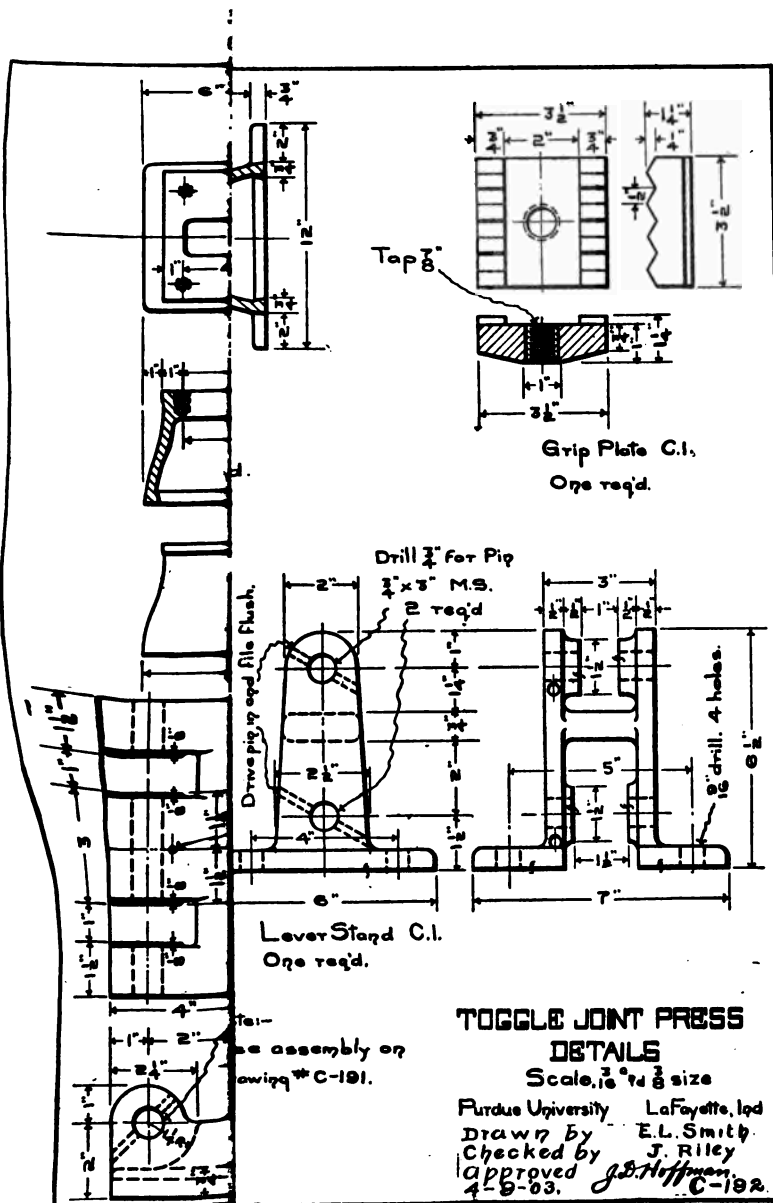
$$(\text{Section at } c) 100 \times 56 = 8,000 h^2 \div 6; h = 2.05 \text{ inches.}$$

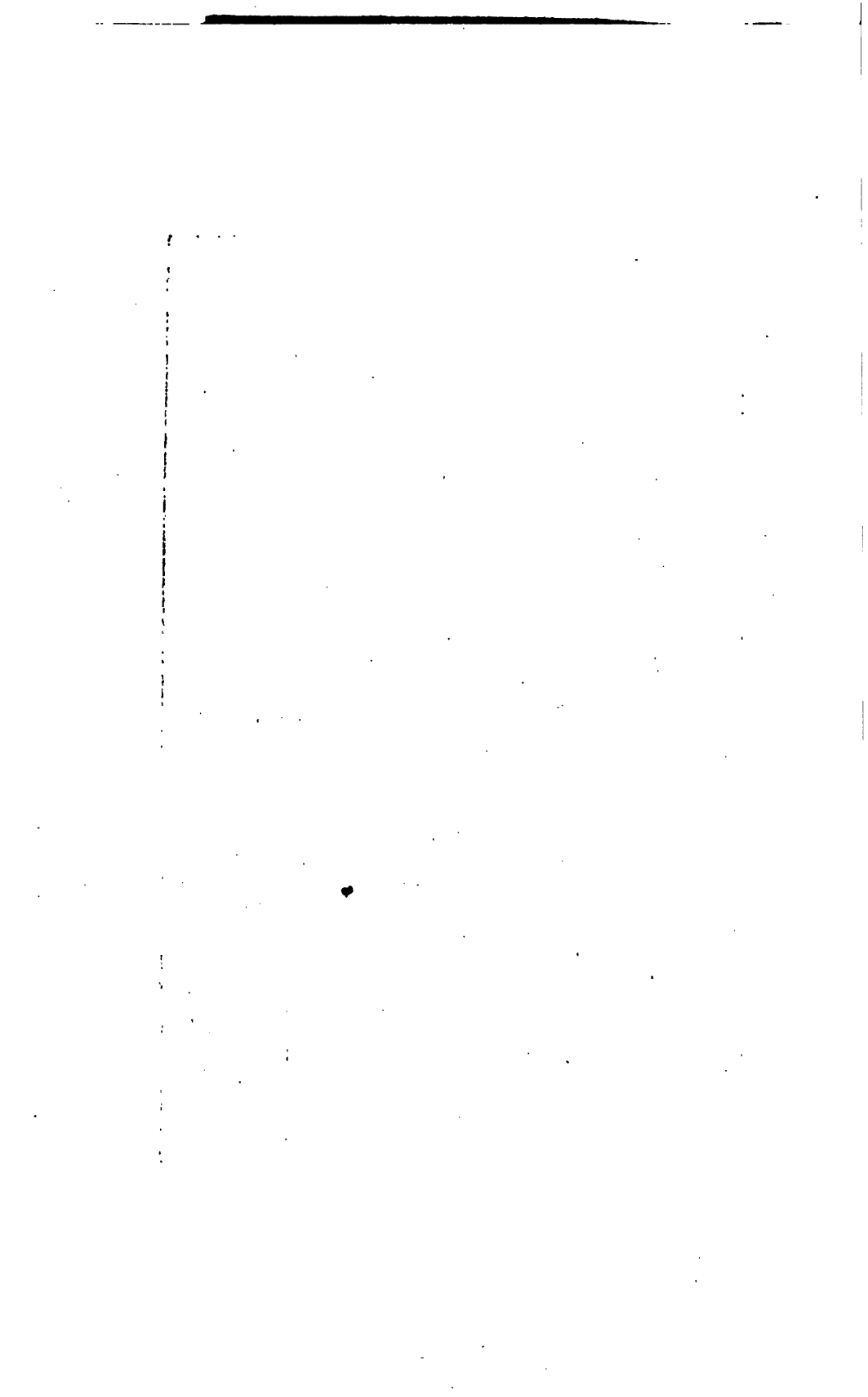
This beam would have a better shape and would also be lighter if the thickness be reduced below 1 inch, say to $\frac{3}{4}$ inch. With this value the formula becomes

$$(\text{At } g) 100 \times 6 = 8,000 h^2 \div 8; h = .77 \text{ inch.}$$

$$(\text{At } c) 100 \times 56 = 8,000 h^2 \div 8; h = 2.37 \text{ inches.}$$

These values give a very good shaped beam, having a section .75 inch \times .77 inch at g and .75 inch \times 2.37 inches at c .





On the other hand, suppose a ratio of b to $h = \frac{1}{4}$, to be desired, the problem becomes

$$(\text{At } g) 100 \times 6 = 8,000 h^3 \div 24; h = 1.22 \text{ inches}$$

$$\text{and } b = 1.22 \div 4 = .3 \text{ inch.}$$

$$(\text{At } c) 100 \times 56 = 8,000 h^3 \div 24; h = 2.56 \text{ inches}$$

$$\text{and } b = 2.56 \div 4 = .64 \text{ inches.}$$

$$\text{section at } g = .3 \text{ inch} \times 1.22 \text{ inches.}$$

$$\text{section at } c = .64 \text{ inch} \times 2.56 \text{ inches.}$$

The above gives the method of determining the size of the section at any point of the beam. Sections should be taken at regular intervals of length and a diagram plotted from the results. One section only need be taken between a and c , say at o midway between. This diagram when completed will show the beam to take the form of a curve similar to Fig. 85. It may be found convenient however, to approximate this curve with a straight line as xy . This would be satisfactory for strength and would be more easily constructed.

It will be noticed that the bending moment becomes zero at the points a and b where the loads are applied. This would theoretically give no size to the handles and make it impossible of construction.

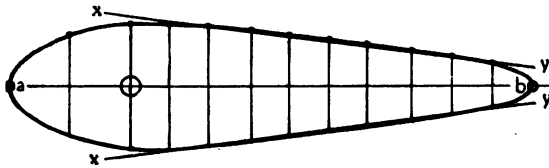


FIG. 85.

Some satisfactory design of handle or hub must be made at these points with sufficient size to carry the pins or bolts, each hub to have the sides and edges of the beam filleted into it in a neat manner. See Plate C-193. A handle can be placed at b for all loads of 300 pounds or less and a drilled hub for larger loads so that a small air or steam cylinder can be attached. A similar hub will be added at a , for connection to the post at the rear.

178. The following shapes may be found useful in designing the lever.

Shapes at b :—The size and shape of the handle or hub at this end will be largely a question of neatness, since the load carried is very small. The pin, if one is used, may be calculated for double shear to get the minimum size allowable, but this size will probably be so small that it will be necessary to increase the size of both pin and hub to add symmetry to the design. Such points as this call for special investigation. Any piece of a machine may be made extra strong, if necessary to harmonize with the other parts of the machine, but the reverse is not the case.

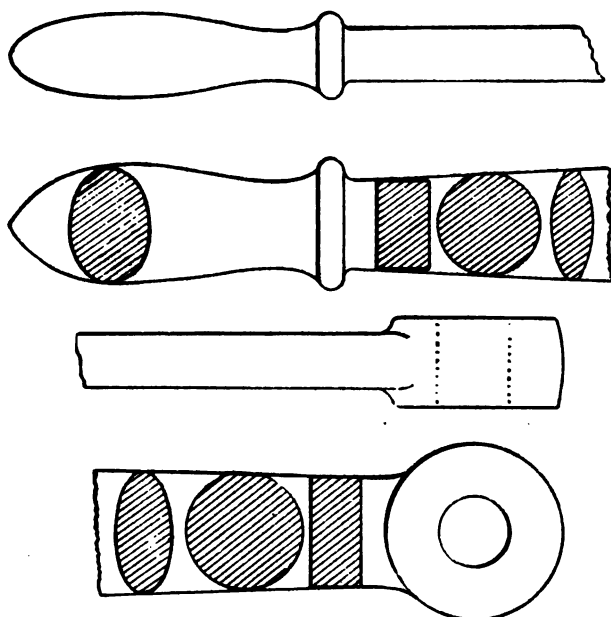


FIG. 86.

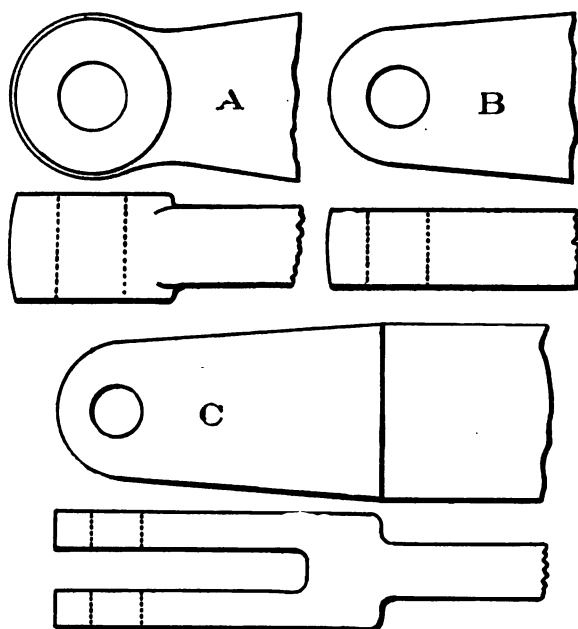
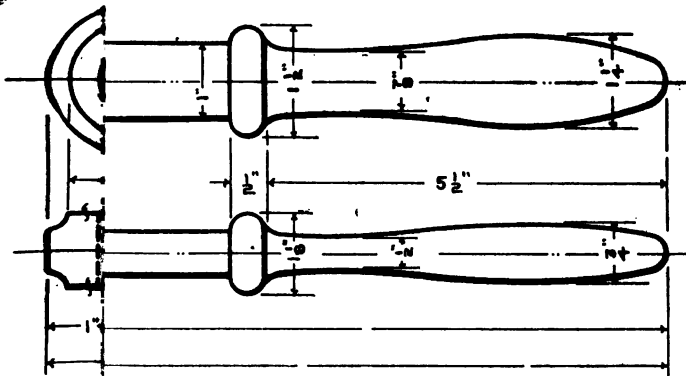
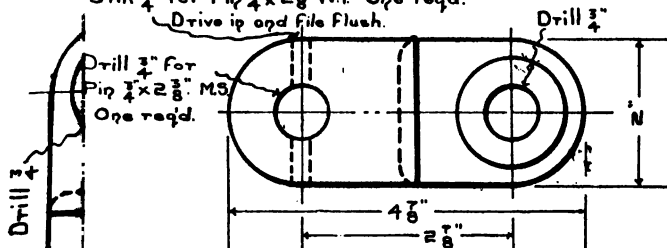


FIG. 87.



Drill $\frac{1}{4}$ " for Pin $\frac{1}{4}$ " x $2\frac{1}{8}$ " W.I. One req'd.
Drive in and File Flush.



Drill $\frac{3}{4}$ " for
Pin $\frac{1}{4}$ " x $2\frac{1}{8}$ " M.S.
One req'd.

Drill $\frac{3}{4}$ "

Assembly
C-191.

Connecting Link. W.I.
One req'd.

TOGGLE JOINT PRESS **DETAILS** Scale, $\frac{3}{4}$ size.

One $\frac{3}{4}$ " bolt, $5\frac{7}{8}$ " long,
over, threaded 1" with two
faced off to a thickness of

Purdue University LaFayette, Ind.
Drawn by E.L. Smith.
Checked by J. Riley.
Approved. *J.D. Hoffman*
4-9-03. C-193.

Shapes at a:—In the construction of this end, Fig. 87, shapes *A* and *B* would be preferred. In most cases the post which connects with it would be made of cast iron and could easily be cored out to fit over the lever arm end rather than to fit the arm end over the post as at *C*. The only calculations necessary here besides figuring the pin, are those that determine the diameter of the hub and the

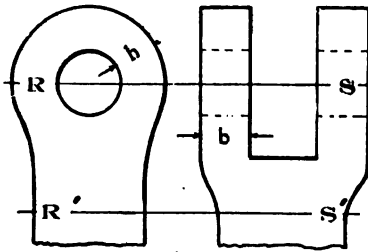


FIG. 88.

length of the hub. It is reasonable to assume that the diameter of this hub should be made equal to the diameter of the cast hub of the standard. To illustrate: at *a*, a tensional force of W'' is acting and this force is resisted by four areas on the section, R S , equal in total area to R' S' , of the standard. These sections are figured for cast iron in direct tension by the formula $W = f A$. The four areas on R S are alike and the ratio of b to h may be assumed. Having figured the pin for double shear by the formula $W'' = 2 f A$, find the diameter of the pin and add to it $2 h$, this will give the diameter of the cast hub and consequently the diameter of the lever end. If f for shear in wrought iron be taken at 5000 pounds per square inch, the diameter of the pin will be .33 inch or, say $\frac{3}{8}$ inch. If f for tension in cast iron be taken at 1500 pounds per square inch, the area $b h$ will be .133 square inch, from which, if b be taken at $\frac{1}{4}$ inch, h becomes .53 inch. This would make the diameter of the hubs at *a* $1\frac{3}{8}$ inches.

It will be next in order to find the length of the hub at the lever end, also the corresponding values of the standard top. These are determined largely from the crushing of the pin. First examine b of the standard to see if the assumed $\frac{1}{4}$ inch is sufficient. The part of the pin in the casting and the part in the lever are both subjected to a crushing force. The resistance of the pin to crushing is in proportion to the projected area of that part of the pin involved; this for the casting is $2 b d = 2 \times \frac{1}{4} \times \frac{3}{8} = \frac{3}{16}$ square inch. Now if f for crushing, written f_c , = f for shear, written f_s , = 5000 pounds per square inch, the pin will sustain a load of $\frac{3}{16} \times 5000 = 938$ pounds safely. This we find is greater than the load W'' actually pulling on the standard so that part of the pin within the cast iron standard is safe. If it had been found that $2 b d$ was so small that the load it was capable of sustaining before crushing was less than the load applied then either b or d or both would be increased. If d were increased without changing b then the hub diameter would be increased this amount above the calculated size of $1\frac{3}{8}$ inches, but if b were increased, the areas $b h$ would be stronger than the

calculated value and h could be reduced accordingly, if it were considered necessary.

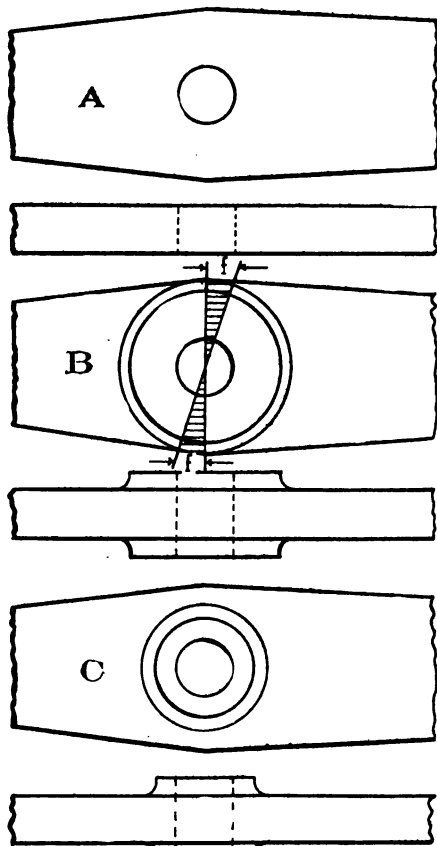


FIG. 89.

obtain the relative resistance offered by the metal at the center as compared to that at the edge of the section. The loss at the center is more than taken up by the addition of a fraction of an inch at the edge or a very small boss around the hole. If the hole in any case should be large, a modulus could be selected for this hollow section, and the exact sizes obtained.

The pin would be calculated in double shear.

The size of the boss, if any be added, is largely optional and is put on for finish.

By the same line of reasoning the length of the pin within the lever would determine the minimum length of the lever hub. This would be $2b = \frac{1}{2}$ inch. From inspection, however, it is readily seen that the thickness at a be necessarily increased to that of the lever section. This at c is .64 inch.

In every fastening of this kind, investigation may be made for *shearing of the pin*, the *tension* on the sections *around the pin*, and the *crushing of the pin*, within both lever and standard.

179. In calculating the size of the section at c the hole was not considered. The error introduced by this is very slight and in most cases may be neglected. The fibre stress in the cross section of the arm varies from zero at the center to a maximum at the edge as shown in diagram B, Fig. 89, where by proportion we can readily

180. Standard Fastening.—In deciding upon the kind of fastening between the standard and the bed, it would be well to first examine it regarding the turning moments about *a*, Fig. 90, where

$W'' b + W_3 h_3 - W'_4 b' = W_a l' + W_y l''$. Assume $b = b' = 3$ inches, $h_3 = 2$ inches, $l' = 5$ inches and $l'' = 1$ inch then with $W'' = 800$, $W_3 = 2552$, and $W'_4 = 450$ pounds. We have $5 W_a + W_y = 6154$.

If $W_a = W_y$ then $6 W_a = 6154$ or $W_a = 1026$ pounds. This is equivalent to a $\frac{1}{2}$ inch bolt. Suppose W_y , because of its location, to be of little value in resisting turning about *a*, then $5 W_a = 6154$ and $W_a = 1231$ pounds = approx. $\frac{1}{8}$ inch bolt. If more than one bolt is used along the line W_x or W_y then the total bolt area may be the equivalent of that given above.

Next examine the joint for a summation of all vertical forces.

$W'' - W'_4 = W_x + W_y$. If $W_a = W_y$ then, $2 W_x = 800 - 450 = 350$ pounds then $W_x = 175$ pounds.

Since this force is less than that obtained by moments it need not be considered.

Next examine the joint for a summation of the horizontal forces. In this the force W_3 tends to shear the bolts off in a plane with the top of the bed, it also acts upon the flanges to shear the casting inside the bolt holes. Considering the bolts first

$W_3 = f A$. If we take $f = 5000$, then

$2552 = 5000 A$, then $A = .51$ square inch of bolt area.

If the bolt shears at the root of the thread, as would be the case with a cap screw, we have at least four $\frac{1}{2}$ inch cap screws needed.

In the second case, if the flange is, say, 6 inches long, we have

$2552 = 2 \times 6 t f$. Let $f = 1500$ for cast iron and we have $t = \text{approx. } .15$ inch.

This would, of course, be made thicker, say $\frac{1}{2}$ inch, for the appearance and good proportion of the casting.

In the above discussion of the standard fastening, the part most liable to fail would apparently be the shearing of the bolts. This might not be true in every case; for example, if h_3 were very great when compared to l' , the failure of the joint would probably be by moments about *a*. The above calculations would be modified, also by the arrangement of the bolts or cap screws.

It is well in every case to examine a joint from all standpoints and design for the greatest requirement.

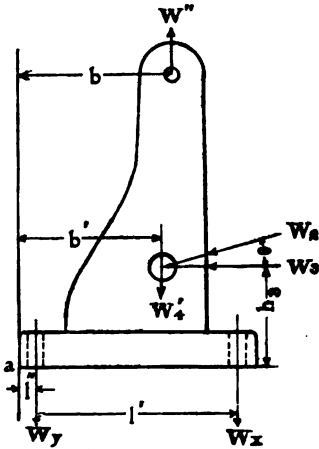


FIG. 90.

181. Standard:—The design of the standard would depend largely on the magnitude of the force to be resisted. In the smaller machines it would undoubtedly be made of cast iron and as such the upper end would be as shown in the preceding paragraph. In the larger machines the standard would be made of wrought iron or steel plates, in which case the sizes of the standard and lever end would be calculated from different values of f than those used for cast iron.

The cross section of the body of the standard may be shaped as in Fig. 91. Assuming the areas to be equal, the strongest section to resist any bending action that may come upon it, is *D*.

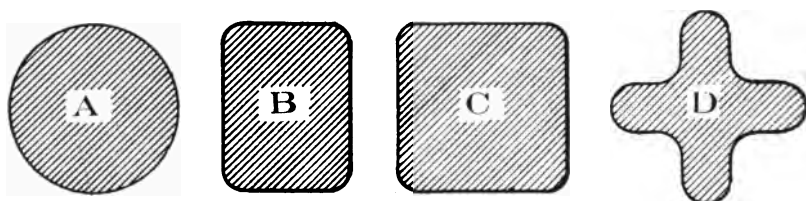


FIG. 91.

The lower end of the standard would be planned to receive the rod W_2 , and would have a flange for fastening to the top of the bed. Fig. 92 shows some of the shapes that may be used.

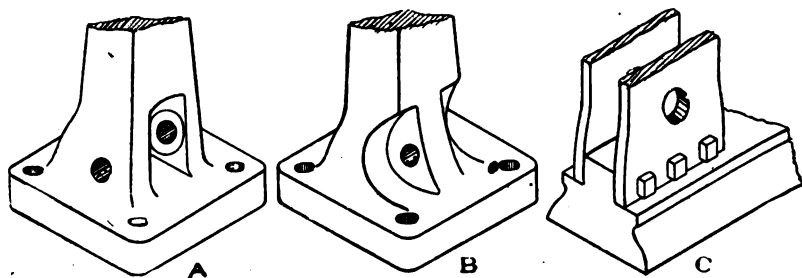


FIG. 92.

The pin at the base is figured for double shear by the formula $W_2 = 2 f A$.

182. Toggle:—There are three ways in which the toggle may fail at the joint: by shearing the pin, by bending the pin and by crushing the pin. In Fig. 93 (A) and (B) show a very simple arrangement of this joint. To obtain the size of the pin from shear in this case,

$W_1 = W_2 = 2 f A$. If $f = 5000$, then $2591.4 = 2 \times 5000 A$; $A = .26$ square inch and $d = .58$ say $\frac{5}{8}$ inch.

It is readily seen that the pin would be found to be the same size if the load $W_1 \div 2$ were figured for single shear.

To obtain the size of the pin to resist bending assume some length of pin between the outer forces $W_1 \div 2$, as 2 inches, and solve by the formula $W' l \div 8 = f Z$. If $f = 8000$, then $900 \times 2 \div 8 = 8,000 \times \pi d^3 \div 32$, then $d = .65$ say $\frac{1}{4}$ inch. There might be a question raised here concerning the proper formula to use for the bending moment, i. e., fixed ends or free ends. The former was selected although the conditions are not well fulfilled.

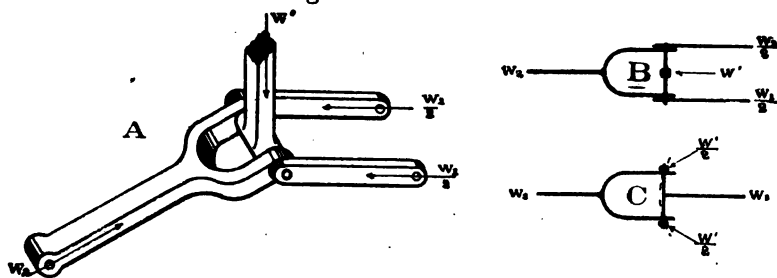


FIG. 93.

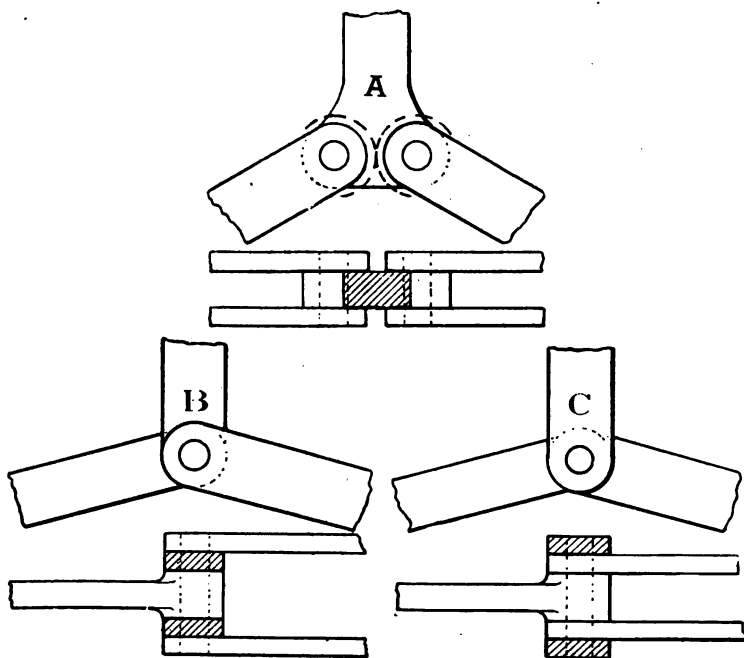


FIG. 94.

The next point to be thought of is the bending of the pin. With the toggle acting on one pin at the center as shown, the smaller force W' should come at the center of the length of the pin as shown in A and B. If the heavier force W_1 or W_2 act at the center of the pin it would have an unnecessary bending strain as shown in C, and would require too large a pin.

Fig. 94 shows other methods of designing the toggle.

Concerning the crushing of the pin see par. 178.

183. Fig. 95 gives some shapes of toggle members; A, B, C, and D are usual shapes of the horizontal members. A and B have split ends and are necessarily hard to forge and machine. C is the simplest form. This form is sometimes modified by adding bosses to one or both sides as shown in D. The vertical member may be constructed solid as at E or adjustable as at F.

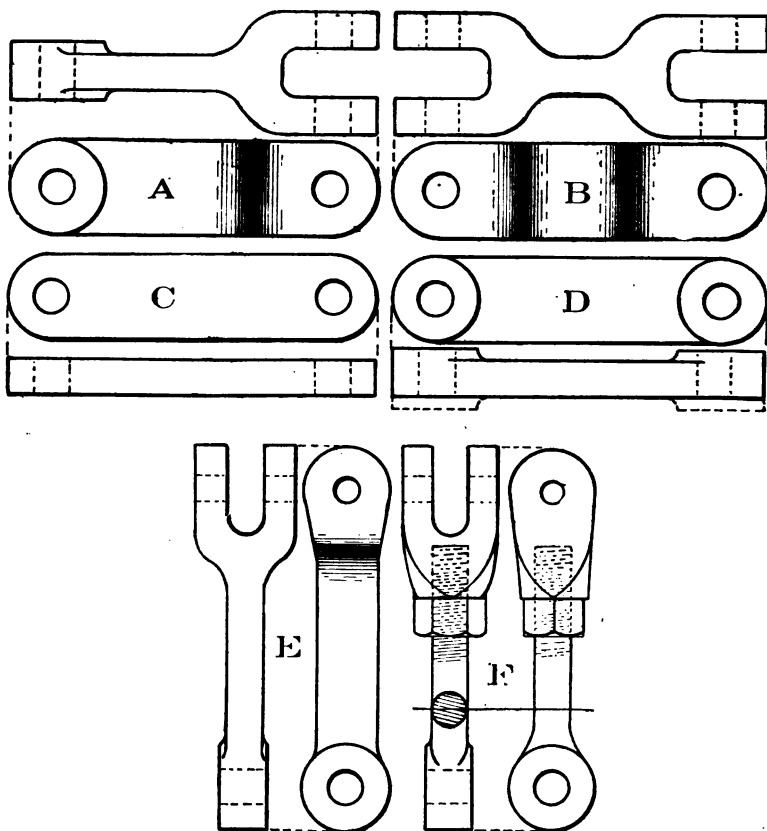


FIG. 95.

184. Die Heads:—The following shapes, Fig. 96, may be of value in designing the sliding head and stationary block. *A* shows the simplest arrangement for fastening these heads to the bed. The sliding head would be made as at *E* for fastening to the toggle and the stationary block would be bolted to the bed as shown at *A*. In such a design the overlap below the top of the bed should be made sufficiently strong to resist the turning action from W_3 . *B* and *C* show the application of gibs between the sliding head and the bed to take up side slack. In some classes of machines such an arrangement would be essential. If, however, heavy side thrusts were involved the form *C* would be questionable unless made very heavy and strong. With the bed planed to an angle as at *C* and *D*, the latter would be considered the stronger.

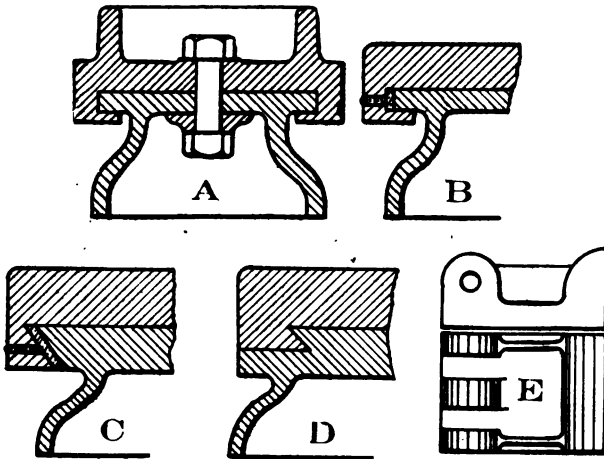


FIG. 96.

185. Sliding Head:—Since the sliding head can not be rigidly fastened to the bed, it must be fitted to a set of guides. The most common fastening is shown in Fig. 97. Having the forces W_3 and W_4 acting on the pin and allowing all the reaction from the die to fall at the upper point of the head, say 4 inches above the bed, we have a cantilever beam acted upon by three forces and tending to break at some section as *a b*. Taking W_3 in two moments about *a*, and W_4 in direct pressure we have $W_4 = f_p A$ and $M = f_m Z$, from which we obtain $f_p = \frac{450}{2 b h}$ due to direct pressure and $f_m = \frac{15312}{b h^2}$ due to the summation of the moments. Now if $f_m - f_p = f_t$; also if $h = 5$ inches and $f_t = 1500$ we have $b = \text{approx. } \frac{3}{8} \text{ inch}$.

If the fibre strength of tension and shear in cast iron be taken the same, then $b' = b$ approx.

In like manner the reaction W_3 from the die may be taken at the bottom instead of the top of the sliding head, and the turning moment figured in this way to see if there is greater danger to the section than when taken at the top.

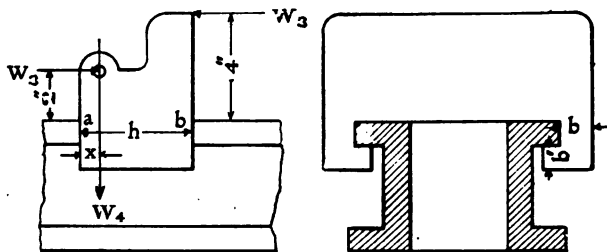


FIG. 97.

186. Stationary Head:—The fastening of the stationary block to the frame to resist the force of W_3 in large machines will call forth extreme care on the part of the designer. The simplest fastening is shown in Fig 98. Take $W_3 = 2552$ pounds, $y = 4$ inches, and $x = 6$ inches, and we have by moments, disregarding the benefit obtained from the overlap of the block around the frame, $2552 \times 4 = 6 W_t$, or, $W_t = 1702$ pounds. It is now necessary to determine if the blocks will slip with this force binding

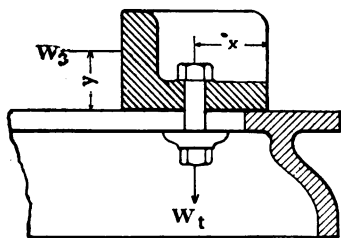


FIG. 98.

the frame between these two friction surfaces. Let the coefficient of friction between the block and the frame also between the washer and the frame be $\phi = \text{say } .3$, then the resistance due to friction is, by formula, $2 \phi W_t = F$ and when applied to our problem is 1021.2 pounds. That is, with the conditions as stated, if W_3 were only 40 per cent as large as it now is the block would just slip. Since the bolt as figured from moments proves to be too small to keep the block from slipping, let us reverse the process and find how large a bolt will be necessary to hold the block against the force W_3 . By substituting as above we have $2 \times .3 \times W_t = 2552$, from which $W_t = 4253.5$ pounds. This force is being exerted at the root of the thread tending to elongate the bolt. With $f = 8000$, this will give slightly greater than .5 square inch of area and will require a bolt of approximately 1 inch diameter. It is evident from this that more than one bolt should be used, or that some other arrangement be substituted for the friction surfaces. In Fig. 99, A, is very similar to Fig. 98, excepting that

the lower surface is notched to protect it from slipping. The upper block may slip slightly, but this will cause a greater grip and a consequent increase of frictional resistance. A possible improvement on this, if the construction of the machine would permit it, would be to have the bolt at an angle as shown in the dotted lines.

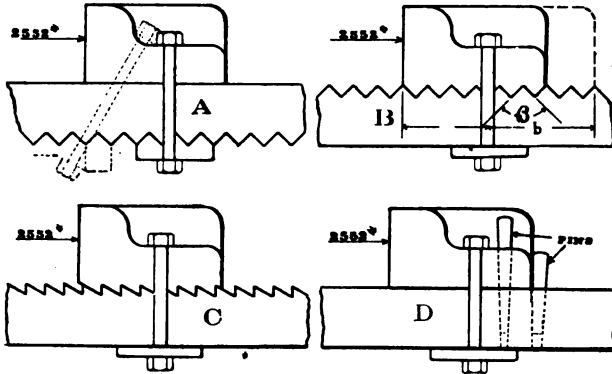
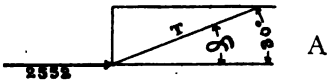


FIG. 99.

Let this angle be, say 30° with the horizontal, then from Fig. 100, A,



A

$.3 T \sin a =$ resistance due to friction.

and $T \cos a =$ horizontal component of the bolt tension, combining

$$.866 T + .3 \times .5 T = 2552, \text{ or,}$$

$$T = 2512 \text{ pounds.}$$

B

This will require a $\frac{1}{8}$ inch bolt.

Fig. 100, B, will cause a tension on the bolt (disregarding friction) of $T' = 2552 \tan (\beta \div 2)$.

Let $\beta = 90^\circ$ then $T' = 2552$

pounds, requiring a $\frac{1}{8}$ inch bolt. It is very evident that if friction were included in this it would reduce the bolt size somewhat.

C, Fig. 99 is probably not as strong in the shape of the tooth as A and B, but with a large tooth area the unit shear becomes small enough so that the teeth are not endangered. The vertical faces on the teeth reduce the vertical thrust on the bolt to a minimum and permit the use of a bolt just sufficiently strong to protect the block from turning as in Fig. 98.

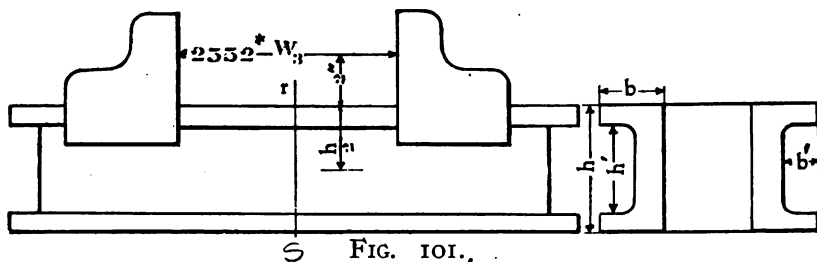
D is arranged to have pins to fasten into the frame either through the block, or behind it. These pins keep the block from sliding and are calculated for shear, while the bolt is calculated to resist turning as in Fig. 98.

Another way in which these fastenings may fail is by shearing the bolt. Assume W_3 Fig. 98 entirely acting to shear, we have

$\frac{2552}{f} = \text{say } 5,000 = .51 \text{ square inches of bolt area.}$ If this is taken as the full area of the bolt it would be $\frac{1}{8}$ inch diameter. This shows a requirement about equal to those for tension. In any form of fastening it is well to investigate both tension and shear and take the larger requirement.

It should also be understood that, if the block clamps over the edges of the frame on planed ways, this will assist the bolt in holding the block down and a smaller bolt may be used.

187. Frame or Bed:—The calculations for the frame will be found somewhat more complicated. Assume a simple type say, of the same general shape and cross section as Fig. 101. Assume also the force W_3 acting at some point along the block face, say at the



middle of the block, a distance of 2 inches above the top of the frame. This force W_3 tends to break the bed along some line as $r s$, and causes a combined bending and tensional stress in the fibres of the section. Considering the part to the right of the section as free we have, Fig. 102, the fibres on the upper or weakside subjected to two tensional stresses the sum of which should not exceed the safe fibre stress of the metal, i. e., $f_1 + f_2 = f_t$; and the fibres on the lower side, subjected to a tensional and a compressional stress, the algebraic sum of which should not exceed the safe compressional fibre stress of the metal, i. e., $f_1 + (-f_2) = f_c$ where

f_1 = uniform tensional stress

f_2 = stress due to bending

f_t and f_c = combined stresses.

To obtain f_1 and f_2 on the tension side use $W_3 = f$, A and $M = f_2 Z$ and obtain

$$\frac{W_3}{A} = f_1 \text{ where } A = \text{area of section in square inches and}$$

$$\frac{W_3 (h \div 2 + 2)}{Z} = f_2 \text{ where } Z = \text{modulus of section.}$$

Having selected the section of the bed as Fig. 101, we find the modulus to be

$$Z = \frac{b h^3 - b' h'^3}{6 h} \times 2$$

It will be necessary here to select some values for b , b' , h and h' and make a trial solution. Take $b = 2$ inches; $b' = 1\frac{1}{2}$ inches; $h = 6$ inches and $h' = 4$ inches.

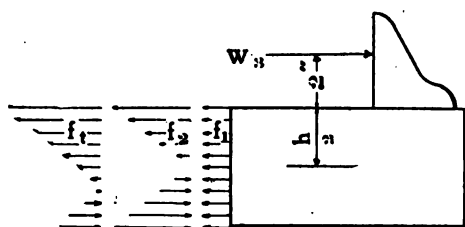


FIG. 102.

Substituting these values in the above we have $Z = 18.8$

and $\frac{2552(3+2)}{18.8} = 679$ pounds per square inch $= f_2$

With the value of b , b' , h and h' as given, the area of the entire section becomes $A = 12$ square inches and

$$\frac{2552}{12} = 212.7 \text{ pounds per square inch}$$

then $f_1 + f_2 = 679 + 212.7 = 891.7$ pounds per square inch.

Since the usual value of f_t for cast iron is 1500 to 2000 this shape and size of section would be stronger than necessary.

Now, if the figures of the section be changed to read $b = 2$ inches; $b' = 1\frac{1}{2}$ inches; $h = 5$ inches and $h' = 4$ inches the value becomes

$$f_1 = \frac{2552}{8} = 319 \text{ pounds per square inch, and } f_2 = \frac{2552(2\frac{1}{2} + 2)}{10.2} = 1126 \text{ pounds per square inch.}$$

$$f_1 + f_2 = f_t = 319 + 1126 = 1445 \text{ pounds per square inch.}$$

This seems to agree very well with the safe value of cast iron in tension, and may be used. Since this is a symmetrical section and since cast iron is much weaker in tension than in compression, the latter will not need to be investigated and the above figures can be accepted for the size of bed. With a section that was not symmetri-

cal it would be necessary to investigate both sides of the section. See par. 196.

Having found the shape of the simple section it is possible to modify it to a certain degree without affecting the calculations seriously. To illustrate, the portion $a b c d$ Fig. 103, may be lopped off and added to the inner side at $a' b' c' d'$ without affecting the modulus. Fillets may then be added at the interior corners giving a shape similar to most frame tops.

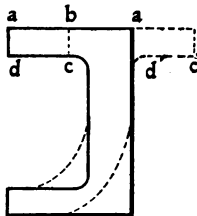


FIG. 103.

For the bottom, a slight deflection or slope of the web can be made as shown by the dotted lines and the result is very similar to a plain cast iron engine or lathe bed. Other minor changes such as slight curves instead of straight sides might be made without any loss of rigidity. In any case where the shape

of the simple section is found and the designer wishes to increase the thickness of any part he may do so and the result is merely to increase the factor of safety.

If under very heavy loads it is advisable to specify one or more steel I beams or channels from Cambria, this may be done by making a trial selection of a section and substituting the value of Z and A in the formulas as before. If this value $f_1 + f_2 = f_t = 8000$ to 16,000, the exact value depending upon the rigidity of the beam, the condition is fulfilled as in the case of the cast frame.

188. The final determination to be made in this design is to obtain the length of the frame to prevent overturning when the load is applied. Let W_s Fig. 104, = the weight of the frame, then from the force diagram we have the following moments about the end at b

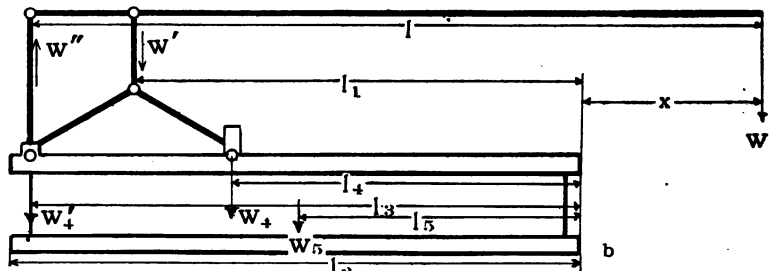


FIG. 104.

$$Wx + W''l_3 - (W_4l_3 + W_4l_4) - W_5l_5 = 0$$

but $W_4l_3 + W_4l_4 = W''l_3$ and $Wx = W_5l_5$.

The length may then be obtained by adjusting the values of x and l_5 such that the equation will be satisfied.

To obtain the length however, in a more direct way the following can be used:

If $x = l - l_3$ and $l_3 = l_2 \div 2$ then $W (l - l_3) = W_s l_2 \div 2$.

Knowing the cross section of the bed in square inches, the weight of one inch in length would be $.26 A$; the total weight of the bed being $.26 l_2 A$ approximately.* Then $l_2^2 = W (l - l_3) \div .13 A$.

Let $l_3 = l_2 - a$ where a is the offset as shown, then $l_2^2 = W (l - l_2 + a) \div .13 A$, from which is obtained the formula

$$l_2 = -3.85 \frac{W}{A} \pm \sqrt{7.7 (l + a) \frac{W}{A} + 14.82 \left(\frac{W}{A}\right)^2}$$

The weight of the frame W_s would be some greater than here shown because of the metal in the ends of the frame and the attached mechanisms, all of which would be effective. The error, whatever it may be, is toward that of safety.

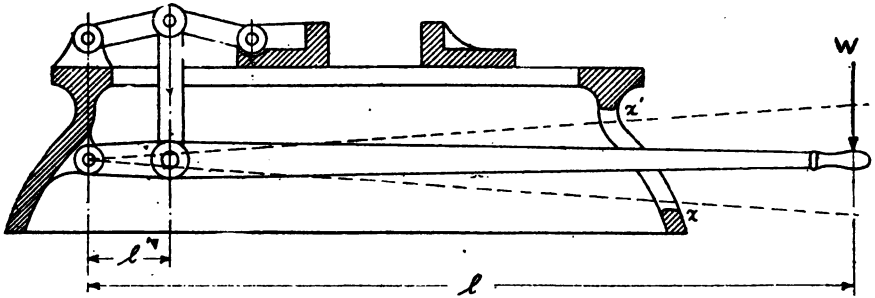
FIRST ALTERNATE DESIGN NO. 1.

FIG. 105, A.

The Toggle Joint Press.**189. Assignment.—**

$$W = \dots; l = \dots; l' = \dots; \theta \text{ (min.)} = \dots$$

In this design the lever is placed within the bed rather than above it. It will be noticed that the end of the bed is slotted to allow for a movement of the lever arm between the points x and x' . The weakening of the bed due to this slot need not be considered a serious matter. With a long and shallow bed however, the movement of the arm will be small and will give a very slight movement to the sliding block. For our purpose this machine may be designed merely to exert a pressure between the two sliding blocks in which case a very slight movement is all that is necessary and the form shown will be satisfactory.

In case the movement of the sliding block is desired greater than that allowed here, the lever may be arranged as shown in Fig. 105, B.

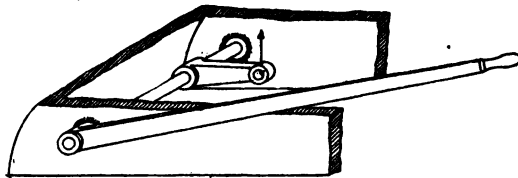


FIG. 105, B.

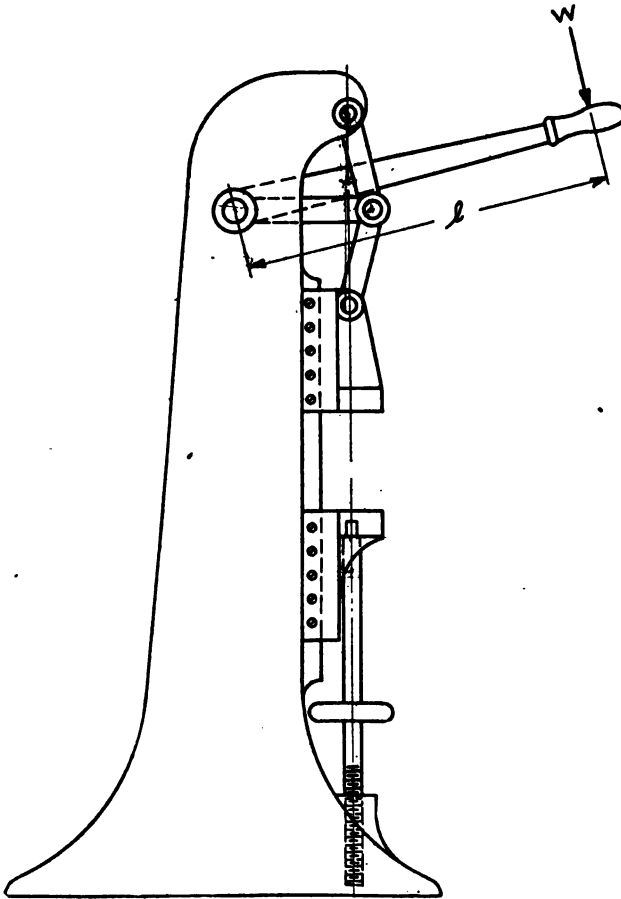
SECOND ALTERNATE, DESIGN NO. 1.

FIG. 106.

Vertical Hand-Power Press.**190. Assignment:—**

$W = \dots$; $l = \dots$; $l' = \dots$; θ (min.) = \dots .

This design follows the principles laid down in No. 1, with two exceptions. First, the length l' here becomes so small that a separate crank can not be used and a bent shaft or an eccentric is substituted. In the eccentric the length l' is the distance between the center of the shaft and the center of the eccentric. Second, the thrust of the sliding block is received through a screw directly against the base of the frame. A hollow rectangular section is suggested as the best shape of the frame. Investigate also for the screw and nut to resist the thrust.

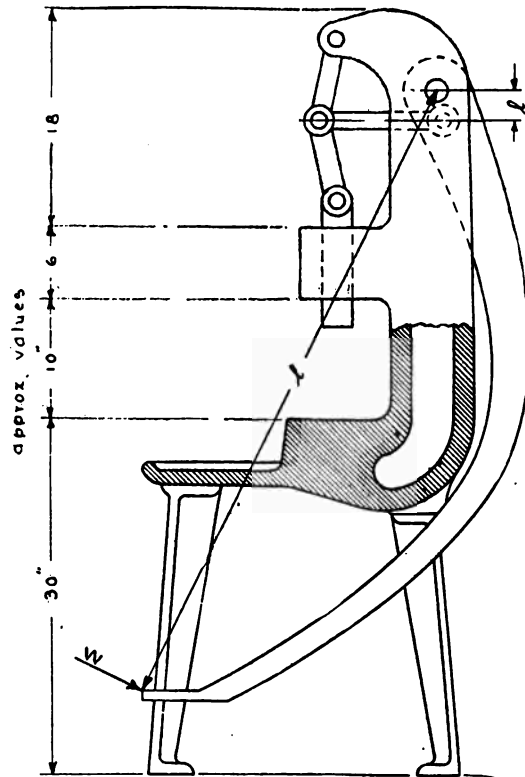
THIRD ALTERNATE, DESIGN NO. 1.

FIG. 107.

The Vertical Foot-Power Press.

(Niles Tool Works Co., Catalog, 1900.)

191. Assignment.—

$W =$ (100 or less).....	pounds
$l =$ (60 to 72).....	inches
$l' =$ (3 to 6).....	inches
θ (min.) =	degrees.

This machine can be used for all kinds of light press work where but a small movement of the ram is needed. Where this movement is desired as great as possible, increase l' and decrease l , also reduce the length of the toggle members.

The ram may be made rectangular in section and the forming dies need not be developed. The frame is hollow and the lever l is fastened on the plane of the toggle.

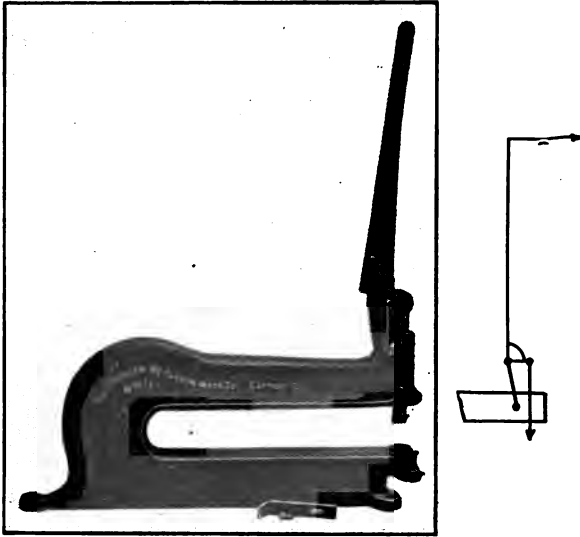
FOURTH ALTERNATE, DESIGN NO. 1.

FIG. 108.

Small Hand Power Punch.

Fig. 108 shows a small bench tool, used for punching sheet iron and other thin metals. Because of its simplicity only two parts of the assignment will be given. All other necessary assumptions may be made by the designer and a complete set of calculations and drawings made. The diagram to the right shows the mechanism.

192. Assignment:—

W (at end of lever l , 50 to 100).....pounds.

T (length of throat).....inches.

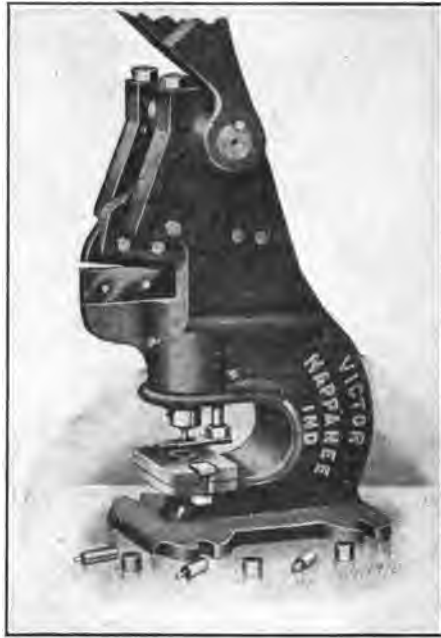
FIFTH ALTERNATE, DESIGN NO. 1.

FIG. 109.

Hand Power Punch and Shear.

The hand power punch and shear is strictly a bench tool for operating on light work. The force at the end of the lever arm l should not be greater than 100 pounds; l' is the eccentricity of the cam, a is the distance from the pivot point of the shear arm to the point where the cam force is applied, and b is the distance from the pivot point to the point of greatest shearing resistance.

193. Assignment:—(See Design No. 2 for details.)

- Kind of material to be cut.....
- Length of cut or diameter of punch.....inches.
- Thickness of plate to be cut (up to $\frac{3}{8}$).....inch.
- Depth of throat.....inches.



CHAPTER VI.

DESIGN NO. 2.

194. General Statement:—A Belt Driven Punch or Shear is the machine selected to represent the second general design. Included within this one machine are problems covering the design of frame, levers, gears, fly wheel, pulleys, bearings, shafts, sliding head, punch, die, clutch, stripper and cam. The fact that this machine finds such general use in manufacturing plants and that it embodies such a variety of designs makes it an ideal subject for analysis. Fig. 110 shows a motor driven shear of late design. It is not expected that the required design will be for a motor drive, but that the distance between the bearings be shortened and pulleys used instead. In giving out the design the following requirements will

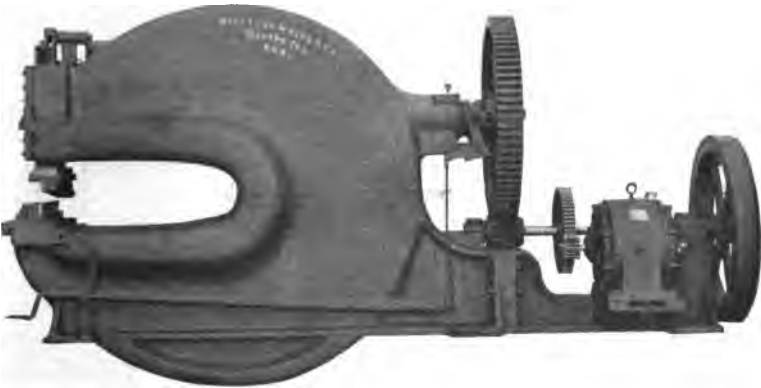


FIG. 110.

be made: First, the work to be accomplished, i. e., diameter and depth of hole to be punched or the cross section of the piece to be sheared; second, the distance from the edge of the plate to the center of the hole, or the depth of the throat of the machine; third, the average cutting velocity of the punch or knife in inches per second, or the R. P. M. of the cam shaft.

In the analysis of the methods employed in working up such a design, the frame sections will be carried out somewhat in detail because of the advanced character of the work; the rest of the machine will be dealt with more briefly. In making the assignments, the members of the class will be given values that differ materially

from those worked out here. The five sample plates at the end of the design show a complete set of drawings of such a machine.

195. Required in the Design:—A machine to punch a one (1) inch hole through three quarters ($\frac{3}{4}$) inch mild steel plate, the center of the hole to be not greater than seven (7) inches from the edge of the plate. The velocity of the punch during cutting may be taken in this case as approximately one (1) inch per second.

196. Frame:—The material used in the frame of such a machine is either close grained cast iron or steel casting. The general shape is about as shown in Fig. 110 and the sections of the frame, Fig. 111, are either hollow cast iron as shown in B and C or web shaped steel as shown in A. Of the three sections, B and C are the most common. Let Fig. 117 represent the outline of the assembly drawing as finally worked out about x, x as the center line of the

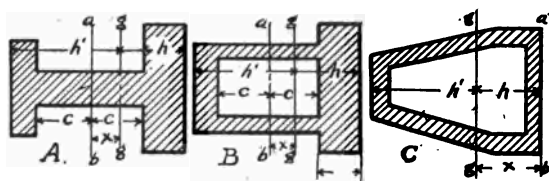


FIG. 111.

the frame. To get the general shape of the frame about the punch, begin by laying off the throat depth, say 8 inches, along the line x, x ; assume some shape of section and figure for the size at several points, then trace the outer curve of the G frame, plan the speed mechanism and locate the shafts. It is necessary many times to modify the first layout a great deal but this must be expected and should not cause discouragement.

To work out the size of the horizontal section along x, x , select the shape, say B, from the standard forms and apply the method used in par. 187, taking G as the depth of the throat and h as the distance from the edge of the casting to the center of gravity of the section.

In applying the formula, $f_2 = \frac{W(h+G)}{Z}$ there may be some confusion in obtaining a satisfactory value for Z owing to the unsymmetrical section. To get Z it will be necessary to determine the *moment of inertia* I of the section and then find Z by the following:

$$\text{for tension } Z = \frac{I}{h} \quad \text{for compression } Z = \frac{I}{h'}$$

Make a trial selection of some sizes for the section and find the neutral axis by cutting out a pasteboard section and balancing it on knife edges, or a better way is by the following: assume any line of

reference as $a b$ Fig. 111, take the algebraic sum of the moments of each rectangular section about this line of reference and divide by the total area; this will give the distance x between the line of reference and the neutral axis $g g$ of the section. When $g g$ is determined find I by the following: To the sum of the products of each area by the square of the distance from its center of gravity line to the gravity axis of the section add the moment of inertia of each section about its own gravity axis. It will be remembered that the moment of inertia of any rectangle about its own gravity axis is $I = b h^3 \div 12$ where h = the total height of the section.

Assume the section with sizes as shown in Fig. 112, then

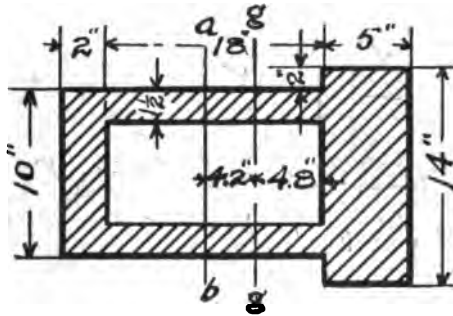


FIG. 112.

$$x = \frac{70 \times 11.5 - 2 \times 10 \times 10}{70 + 20 + 54} = 4.2 \text{ inches}$$

$$I = 70 \times (7.3)^2 + 54 \times (4.2)^2 + 20 \times (14.2)^2 + \frac{14 \times (5)^3}{12} + \frac{3 \times (18)^3}{12} + \frac{10 \times (2)^3}{12} = 10326$$

$$Z_t = \frac{10326}{9.8} = 1054 \text{ for tension}$$

$$Z_o = \frac{10326}{15.2} = 679 \text{ for compression.}$$

The value W is the pressure on the punch and if the ultimate shearing stress of mild steel be taken at 55000 pounds per square inch, would be 129,591 pounds.

Considering the section only on the tension side we have $f_1 + f_2 = f_t = 900 + 2189 = 3089$ pounds per square inch. This fibre stress would be large for cast iron hence another section must be selected.

Take for a second trial the section Fig. 113, we have, if worked as above

$$x = 2.97 \text{ inches}$$

$$I = 21049.44$$

$$Z = \begin{cases} 1680 & \text{for tension} \\ 1317 & \text{for compression.} \end{cases}$$

$$f_1 + f_2 = f_t = 2154 \text{ pounds per square inch.}$$

In like manner we should work out the compression side by $f_1 - f_2 = f_c$. See also par. 187.

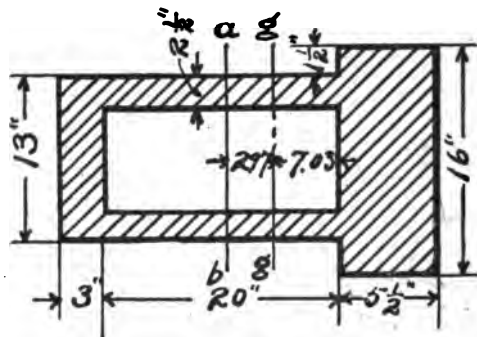


FIG. 113.

Any other section of the frame can be determined by working out f_2 as shown above and combining with it the value of $f_1 = W \cos a \div \text{area}$. The value of f_1 is a maximum when a is zero and becomes zero when a is 90° . It will be seen, Fig. 114, that at section

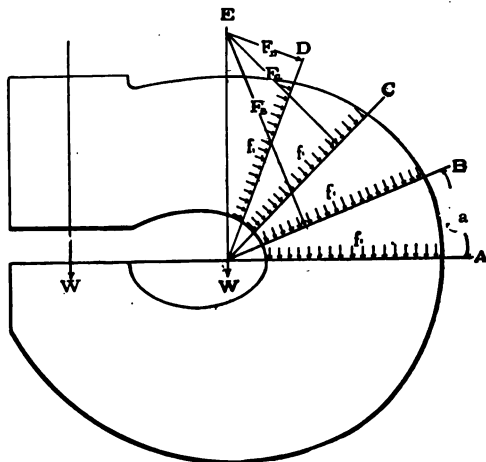


FIG. 114.

$A, f_1 = W \div \text{area } A$; at $B, f_1 = F_b \div \text{area } B$, but $F_b = W \cos a$ hence $f_1 = W \cos a \div \text{area } B$; at $C, f_1 = W \cos a \div \text{area } C$ and so on until f_1 becomes zero at section E . At this point the frame should be examined for both bending and shearing and the larger requirement taken. In all probability section E will be made larger than the calculated size to accommodate the finishing around the head. It will be satisfactory in this design if we obtain sections at $a = 0^\circ, 45^\circ$ and 90° .

To find a section at say $a = 45^\circ$ determine the height of the gap and draw the outline. This shape is controlled by the space taken up by the dies, and the clearance for the metal to be punched. It can not be determined exactly but by Par. 206 a good estimate may be made. Assuming some section of the frame as Fig. 115 and solving for the fibre stress as before, we find

$$x = 2.67 \text{ inches.}$$

$$I = 15655.$$

$$Z_t = 1382.$$

$$f_1 + f_2 = f_t = 1872 \text{ pounds per square inch.}$$

NOTE.—In finding M in $M = f Z$ the lever arm varies approximately with the cosine of the angle a .

Investigate also for f_c .

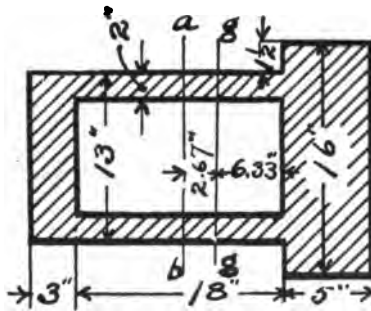


FIG. 115.

For the vertical section take Fig. 116, in which case we have,

$$x = 3.26 \text{ inches.}$$

$$I = 4411.79$$

$$Z_t = 655.$$

$f_2 = 791$ pounds per square inch, which shows that the section could be materially reduced in size if it were desired. This reduction could very properly be made according to the dotted lines. If it were considered necessary, this section should also be investigated for compression.

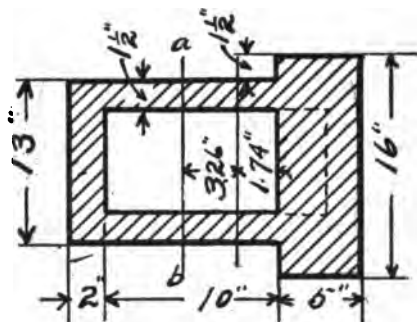


FIG. 116.

To investigate for shearing on the vertical section we have, allowing the shear to be absorbed by the entire section.

$$f_s = \frac{129591}{136} = 953 \text{ pounds per square inch.}$$

197. Having determined several important sections in the frame, the outline can then be drawn in as in Fig. 117. This outline will of course be modified somewhat for the shaft, head and leg.

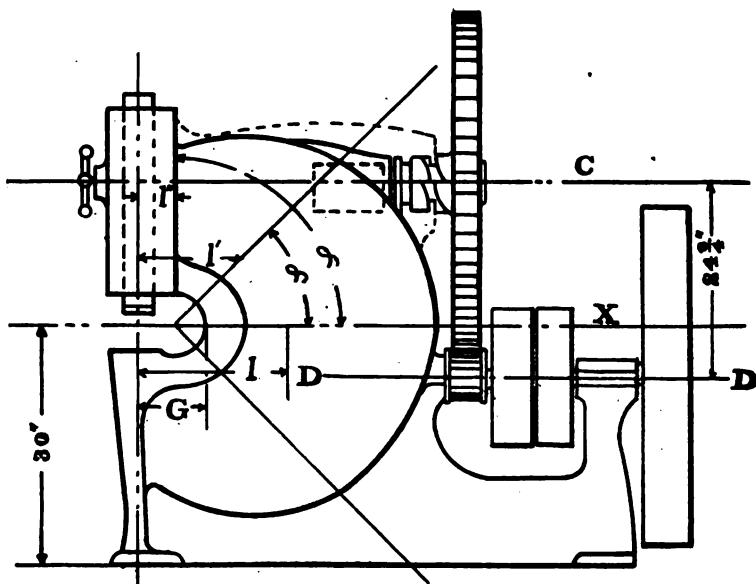


FIG. 117.

It will be noticed that a somewhat higher fibre stress has been allowed in this frame than in the material as used in the toggle frame. This is about as would be expected. Any casting planned to fill a very important place in the design of any machine would be made of the best close grained gray iron. It is advisable to keep the size of this frame as small as possible consistent with strength and, since the best of cast iron would have an ultimate strength of 25,000 to 30,000 pounds per square inch, it would be considered safe to allow a fibre stress of 2,000 to 2,500 pounds per square inch, which corresponds to a factor of safety of 12.

The shape of the section may be varied to suit the conditions, from a large and thin section as here treated, to a small compact and possibly solid section. The latter condition prevails in some machines where the gap is long and the main section would be necessarily crowded into the smallest space.

Steel cast frames are very common, especially on the larger machines. When made of steel the frame section may be made much smaller. $f_t = 12000$ to 15000 pounds per square inch.

Tension bars are provided for machines with long gaps. These bars are very necessary when doing heavy duty.

198. The Maximum Punching or Shearing Force is used in calculating the frame sections. The ultimate shearing stress of the metal multiplied by the area to be cut gives the maximum load on the punch or flat shear. If the greatest load on a *bevel shear* is desired, multiply the maximum load on a flat shear by the following:

THICKNESS OF THE METAL.

	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{7}{8}$	2
4° Bevel	.42	.48	.54	.61	.67	.73	.79	.85	.92	.98	
8° Bevel	.23	.3	.37	.44	.51	.58	.65	.73	.81	.88	.95

Look up articles on the Shearing of Metals in the American Engineer and Railway Journal. Vol. 67, Page 142.

In any machine of this kind it is safe to allow 15 to 20 per cent for the friction of the parts while performing the heaviest duty. The total pressure to be accounted for at the driving end in this machine will then be $129591 \div .85 = 152460$ pounds. If the eccentricity of the cam be taken the same as the thickness of the thickest metal to be punched = $\frac{3}{4}$ inches, the twisting moment from the gear side will be approximately $\frac{3}{4} \times 152460 = 114345$ inch pounds.

199. Working Depth of the Cut:—The actual cutting depth (depth of penetration) of a punch or flat shear is used in determining the foot pounds of work done at the tool, and is a certain *percentage of the total thickness of the metal*. Generally the tool in its movement passes entirely through the metal, but the work of cutting is finished when the tool arrives at the depth of penetration.

This percentage varies somewhat with the kind of the metal, but for mild steel it has been found by experiment (American Machinist, Oct. 12, 1905) to be

Thickness of Metal, in. 1 $\frac{3}{4}$ $\frac{5}{8}$ $\frac{1}{2}$ $\frac{3}{8}$ $\frac{1}{4}$ $\frac{1}{8}$ $\frac{1}{16}$ $\frac{1}{32}$ $\frac{1}{64}$
 Depth of Penetration, .25 .31 .34 .37 .44 .47 .5 .56 .62 .67 .75 .87%

200. Diameter, Width and R. P. M. of the Pulleys:—

Table XXII gives values agreeing fairly well with current practice for the diameter and revolutions per minute of the pulleys. To determine the width of the pulley face, or the width of the belt, no

TABLE XXII.

Machine will Punch	Diameter of Pulley.	R. P. M.
$\frac{1}{4}" \times \frac{1}{4}"$	10	200 to 250
$\frac{1}{2}" \times \frac{1}{2}"$	12	200 to 250
$\frac{3}{8}" \times \frac{3}{4}"$	16	175 to 200
1" x 1"	18	150 to 175
2" x 1"	30	150 to 175

definite rule can be stated. Practice varies between a 2 inch belt on a $\frac{1}{4}$ inch x $\frac{1}{4}$ inch machine, and a 6 inch or 7 inch belt on a 2 inch x 1 inch machine. Calculations for belt sizes on such machines do not give very satisfactory results because of the small percentage of each revolution that the machine is actually working. It is a good experience, however, if each man would apply a few trial conditions and note the results. First find the effective pull P on the belt, by the horse power formula or by moments from the cam shaft, assuming the punch or shear to be cutting full value all the time, and then take the percentage of this which is represented by the proportion of the total time that the cutter is actually working. Figure the belt from this result as in Par. 91. In all probability, catalog sizes will finally be taken.

201. Fly Wheel. Weight:—The weight of the fly wheel may be obtained by either one of two methods; first, by assuming the wheel, when running at full speed, to have stored up energy enough to do a certain definite work; second, that the wheel shall have only a certain allowable fluctuation from full load to no load. From the first method, a fly wheel for a machine of this kind may be designed to fulfill a number of conditions, from a wheel such that its kinetic energy will just equal the energy absorbed by the machine during punching, (in which case if we disregard the belt's action, the velocity of the wheel would become zero after each hole punched), to a wheel of such a size that the residual energy will be

sufficient to keep the speed fairly constant. Current practice approaches the former and in this consideration will be adopted.

Having given the force to be accounted for at the driving end as 152460 pounds, assume that this force acts through say a maximum of one-half the total depth of the cut, which is $\frac{3}{8}$ inch or $\frac{1}{4}$ foot, then the energy exerted would be 4764 foot pounds. Apply the formula $W v^2 \div 2g$ to the mean rim diameter. Assuming 36 inches as this diameter we have:

$$\frac{W v^2}{2g} = 4764; W = 553 \text{ pounds.}$$

The depth of penetration here was not used according to the table. This should not be confusing since it is merely for illustration.

Work out the weight of the fly wheel from the allowable fluctuation par. 118 using say, 20 to 25 per cent, and check with the above.

Arm:—The fly wheel arm may be calculated as follows: Estimate the time required in punching one hole then find the distance through which a point on the center line of the rim will move during this time; this will be the value V in $P V = 4764$. Since there are 15 R. P. M., each revolution will take 4 seconds. Assuming the velocity of the punch during action to be the same as that of the cam center we have $V = 3.1416 \times 1.5 \div 4 = 1.1781$ inches per second. The time occupied in punching is $\frac{3}{8} \div 1.1781 = .318$ second. The velocity of the rim of the wheel is 1413.7 F. P. M. = 23.56 F. P. S., from which we find that the rim will travel 7.5 feet before stopping. Applying $P V = 4764$.

$$P = 635 \text{ pounds.}$$

The value P may be found in another way. Find the force p at the gear, Fig. 118, from the moments around the cam shaft, this is

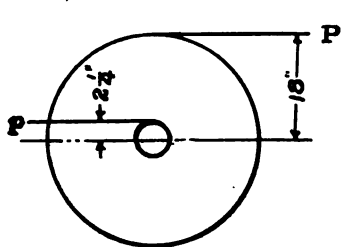


FIG. 118.

$$p = \frac{152460 \times 3}{4 \times 22.5} = 5082 \text{ pounds}$$

then by moments around the driving shaft

$$P = \frac{5082 \times 9}{4 \times 18} = 635 \text{ pounds}$$

Having found P , obtain the large dimension of the arm at the center of the shaft from formula par. 119.

$$b = \sqrt[3]{\frac{P R}{6 \times .05 \times 1500}} = \sqrt[3]{\frac{635 \times 18}{6 \times .05 \times 1500}} =, \text{ say, } 3 \text{ inches.}$$

For details of shapes and sizes of rim, arms and hub see par. 124

202. Driving Shaft:—If the bearings are close to the pulley and gear the bending will not be excessive and the shaft may be figured to resist twisting with a low fibre stress. Taking $f = 6000$, the diameter of the shaft will be 2.2 say $2\frac{1}{4}$ inches.

On machines where the pull of the belt and the side thrust from the gears are fairly great, also when the bearings are far apart, it is necessary to design the shaft for combined twisting and bending. In such a case find the side thrust due to each, the belt and the gears, and calculate from the bending moment as a beam fixed at the ends and loaded at two points. See par. 57.

In locating the shaft D, D, it will first be necessary to have the approximate location of the main shaft and the diameters of the gears. Knowing the angular velocities of the two shafts the diameter of the small gear may be assumed and the distance between the shaft centers obtained. In this machine if the cutting speed of the punch is one inch per second, the center of the cam will travel approximately $60 \div 4.71 = 13$ revolutions per minute. Calling this 15 and the revolutions per minute of the pulley shaft 150 the ratio of the gears is 10. With $4\frac{1}{2}$ inches as the diameter of the pinion, the shafts will be $24\frac{3}{4}$ inches between centers.

Note:—The value $4\frac{1}{2}$ as the diameter of the pinion was taken merely for illustration. This would be rather small for the construction of a perfect tooth.

203. Main Shaft:—The main shaft or "cam shaft" as it is sometimes called would be made of hammered steel. Figs. 119 and 120 show two common forms, B_1 , B_2 and B_3 are journals. C is the cam which operates the punch. The greater part of the thrust from the punch is absorbed at the journal B_2 , B_3 being added for the double purpose of reducing the strain of the shaft and for an outside connection for adjustments.

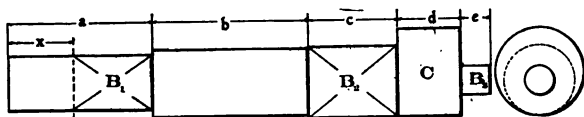


FIG. 119.

In designing the shaft the part a may be figured to resist the twisting moment due to the thrust on the gear x , allowing a fibre stress of, say, 6000 pounds per square inch for shear. It will be noticed, however, that the thrust on the gear produces a bending moment on the shaft, the lever arm being $\frac{x}{2}$. This bending moment

may be of such magnitude as to make it necessary to use the combined formula Par. 68. It would be well to obtain the diameter from both formulas and see if they will check each other.

The length of the journal may be taken from 2 diameters to 2.5 diameters of the shaft. The length of b will be quite variable

and will be governed by the frame of the machine. The diameter of b will depend upon the judgment of the designer; in some shafts it is made equal to the diameter of the left journal while in others it is enlarged to the size of the main journal. A high speed machine would require a larger and stiffer shaft than a slow speed machine, because of the heavy shocks to which the shaft is subjected, hence, the diameter of b would be as large as possible.

Take the size of the main journal such that the pressure per square inch of projected area will not exceed 3000 pounds assuming the entire thrust from the punch to be taken up by it and that of the cam not to exceed 8000 pounds. Lower values than these are desirable, especially on the cam, where 5000 pounds per square inch of projected area is a good value. It will be seen from the

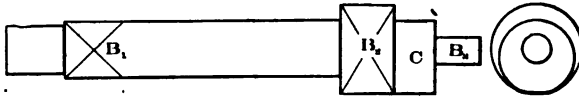


FIG. 120.

above that, the projected area being constant, a bearing may be changed in shape decidedly and yet give good service. As an illustration B_2 may be long and slender as in Fig. 119, or short and thick as in Fig. 120, so long as the shaft at this point is stiff enough to resist bending and shear. Conditions within the machine itself usually determine the shape of bearing and cam. When the sizes are approximately determined then they should be constructed graphically to scale, usually having the two surfaces continuous along one line.

The cam varies from 3 to 6 inches in length, and from 6 to 12 inches in diameter. The diameter of the bearing in such a case is governed somewhat by the eccentricity of the cam.

The cross sectional area of the bearing B_2 along its upper face next the cam must be sufficient to resist the effect of *shear*; it must also resist the *bending moment* produced by the thrust multiplied by the half length of the cam ($\frac{d}{2}$) and the torque produced by the thrust multiplied by the eccentricity of the cam. This should be worked by the combined formula, remembering that B_1 , where used, would reduce this bending moment somewhat.

In machines where the distance between B_1 and B_2 is great there is a bending of the shaft between the bearings. This is especially true where B_3 is omitted as in some horizontal machines. Such a condition is equivalent to a beam in flexure with the reactions at B_1 and C and the applied load at B_2 . The effect, however, is not the same in the calculations as a simple beam because of the support given to it by the boxes.

It is safe to assume that the bearings are sufficiently loose to allow some bending, but not loose enough to consider the maximum

loads as applied at the centers. Probably a safe assumption would be 50 per cent of the maximum load applied at the cam center and resisted at the bearing centers.

The frame should be fitted with a phosphor-bronze bushing $\frac{1}{4}$ inch to $\frac{3}{8}$ inch in thickness surrounding the journal B_2 . This bushing is made a forced fit with the frame.

The sizes of B_3 would vary between 2 inches and 4 inches for both diameter and length.

Application:—Figuring the shaft for twist at its smallest diameter, at the gear, gives $d = 4.59$ say 4.5 inches.

The cam diameter, assuming a length of 4 inches and a pressure per square inch of 5000 pounds is $\frac{129591}{5000 \times 4} = 6.5$ inches.

B_2 will then be 5 inches diameter, and if we allow 2500 pounds per square inch projected area, will have a length of $\frac{129591}{2500 \times 5} = 10.4$ inches, say 11 inches.

B_3 may be taken say $2\frac{1}{2}$ inches long by 3 inches diameter.

204. Sliding Head:—Of the different types in use, two of the very common ones are shown in Figs. 121 and 122, the former being more common in the smaller machines. The chief objection to the bronze block is its liability to wear unevenly thus causing lost motion and an irregular movement of the block while punching.

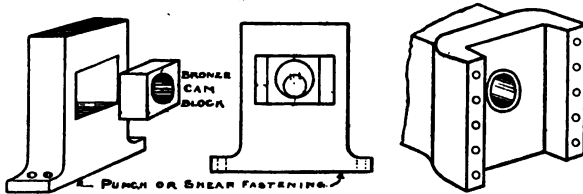


FIG. 121.

In the latter form, the entire thrust is carried on a hardened steel block set into the cast iron sliding head and the wear, if any, is practically uniform. The size of the bearing surface in the steel block may be figured from the crushing strength of the steel casting. If this value be taken

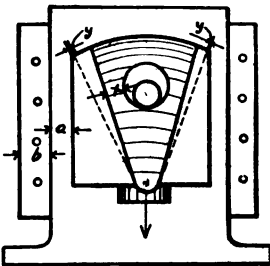


FIG. 122.

at 90,000 pounds per square inch with a factor of safety of 6, the projected area of this bearing will be $129591 \div 15000 = 8.6$ square inches, from which if the length of the cam be 4 inches, the breadth of the bearing will be 2.15 inches, say $2\frac{1}{4}$ inches. The breadth of the sliding head face will be seen to

depend upon the construction of the vibrating arm. Make the vibrating arm a steel casting and allow from $\frac{3}{4}$ inch to $1\frac{1}{4}$ inches at x , and a small clearance at y . This part of the work must be done graphically. The values a and b will depend respectively, upon the width of the frame and the diameter of the bolts used.

205. Clutches and Transmission Device:—In operating any machine having an intermittent motion a clutch is commonly used to serve as a connector between the power supply and the work. The application of the clutch to the simple punching or shearing machine is shown in Fig. 123. It is usually applied directly to the hub of the large gear and is operated through a system of levers and cranks by either hand or foot. When the punch is not operating, the large gear, which is designed with a long hub to act as a bearing, runs loose, the shaft remaining stationary. The clutch sleeve slides on the shaft over a splined key and when the punch is to be operated this sleeve is thrown to engage with the corresponding part on the gear hub. When the hole is punched a counter weight brings the sleeve back to its former position and the movement of the punch ceases.

Clutches are formed with either two, three or four jaws. These jaws may be formed as a part of the wheel hub as shown at A and B cast from steel and bolted to the flat face of the wheel hub as shown at C , or cast from steel and fitted to the interior of the wheel hub as shown at G . In heavy work C and G are preferable.

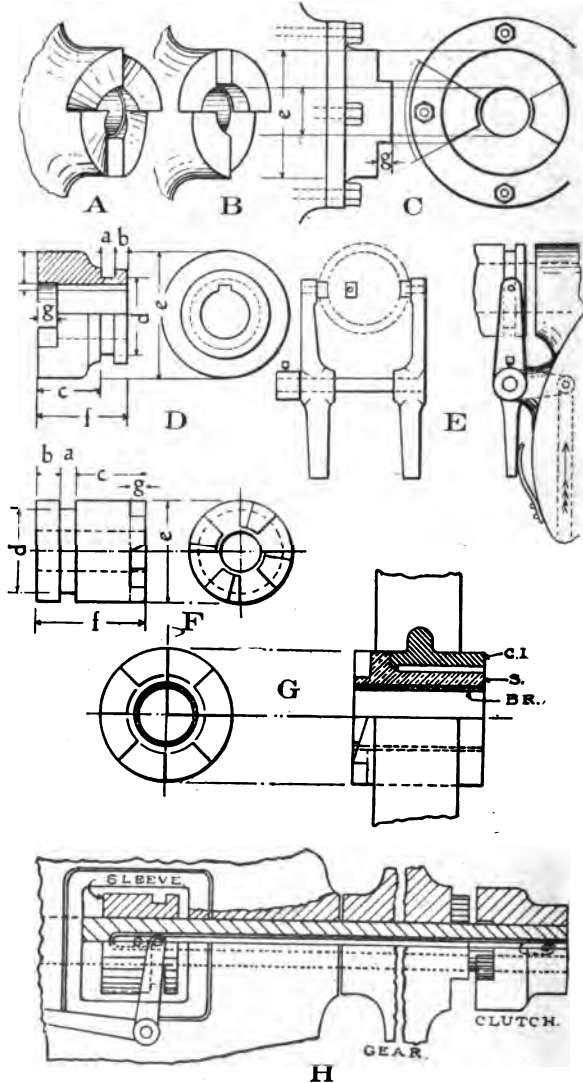
The part of the clutch subjected to the greatest wear is the front face of the jaw. This is sometimes faced with a plate of high carbon steel which can be replaced when necessary with a new one. The rear face of the jaw is usually perpendicular to the front face of the wheel but is sometimes cut to an angle of 30 to 45 degrees. There should be sufficient clearance between the jaws on the sleeve and the wheel to enable them to be easily thrown together while in motion. This should be from $\frac{1}{8}$ inch to $\frac{1}{4}$ inch.

The clutch sleeve may be shaped as shown in either D or F . The following sizes, table XXIII, will meet average requirements.

TABLE XXIII.

Shaft=	2"	3"	4"	5"	6"
a	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$
b	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$
c	= f — (a + b)				
d	4	$5\frac{1}{2}$	7	9	$10\frac{1}{2}$
e	5	7	9	11	12
f	3	4	$5\frac{1}{2}$	7	8
g	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$

There are two general methods of designing the transmission device; the first and simpler one *E* having the clutch between the gear and the frame, and the second *H*, having the gear between the clutch and the frame. The latter method necessitates a hollow shaft in order to obtain a rigid connection between the sleeve and the clutch and is not much used on small machines.



H
FIG. 123.

206. Punch, Die, and Holders:—In all punch and die work the die is made some larger than the punch for clearance. The action of the punch on the material is shown in *A*, Fig. 124; the hole

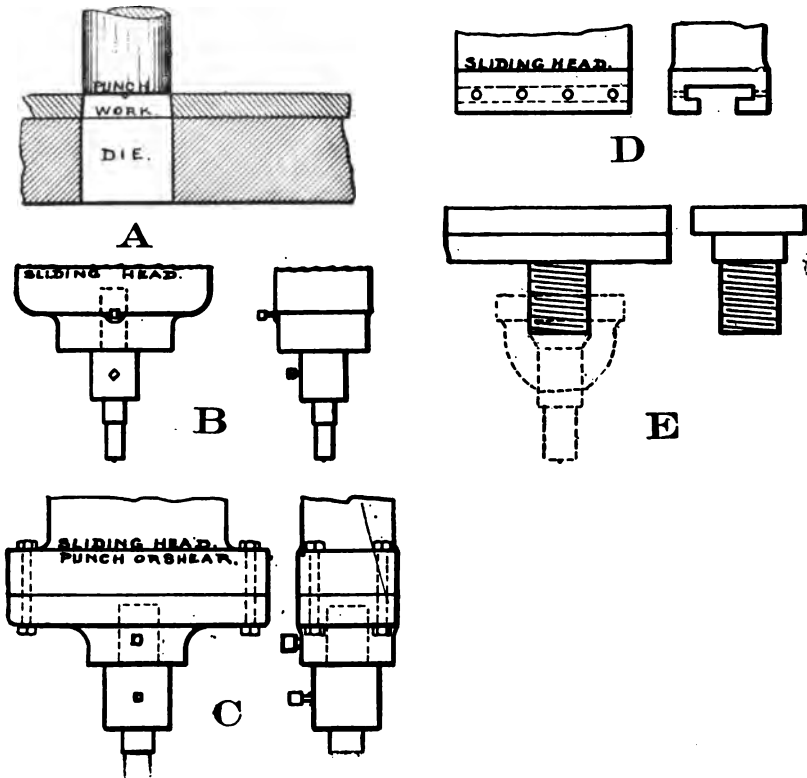


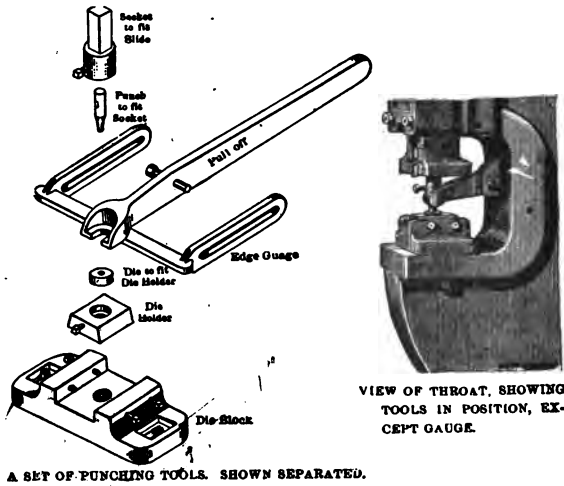
FIG. 124.

tapering, from the size of the punch on one side to the size of the die on the other. This taper is slight and is considered of no consequence in rough work, but in finished work it is a difficulty that can easily be remedied by reaming the hole afterwards. For complete treatise see "Dies, Their Construction and Use." Woodworth.

There are various methods of fastening the punch to the sliding head; *B* shows the bottom of the sliding head fitted with the square ended socket and punch. A screw ended socket is sometimes used instead of the square shanked socket. *C* shows the bottom of the head flanged and drilled for the attachment of either punches or shears. In single machines it is desirable that both punching and shearing be done; where such is the case this is a good

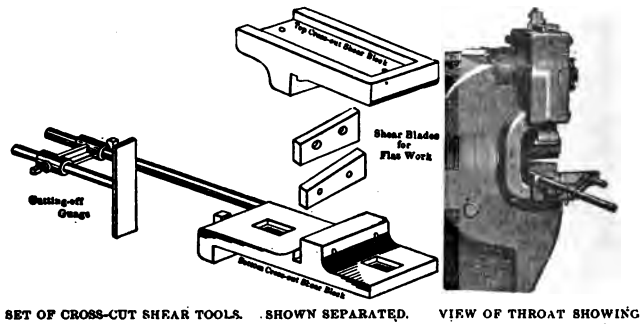
form. Side adjustment of the punch may easily be made if the head be slotted as at *D* and fitted with a tee block as *E*. Dies are made from high carbon steel and are held in a holder; the holder in turn is bolted to the horizontal face of the frame. A certain amount of adjustment is necessary in locating the die consequently the holder is made in two parts.

The application of punches and shears to machines is well shown in Fig. 125.



A SET OF PUNCHING TOOLS. SHOWN SEPARATED.

PUNCHING TOOLS.



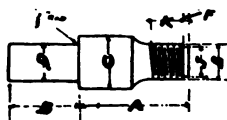
CROSS-CUT SHEARING TOOLS.

FIG. 125.

The type of punching tool in most common use is that shown in Fig. 126. The sizes of the various parts are obtained as a matter of experience rather than from calculation. In order that each designer may have quick reference to accurate data on this part, the following table, kindly furnished by Messrs. Williams, White and Co., Moline, Ills., is added. In referring to this table it must be understood that all punches included under one letter are the same length and fit the same coupling nut. It should also be stated that reducer couplings are sometimes provided so that the same stem may be used for different sized punches.

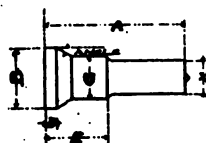
TABLES XXIV.

STEMS.



No. of Coupling.	Diameter of Punch.	Regu- lar Length Punch.	A.	B.	C.	D.	E.	F.	J.	K.
C	1'-1"	1 1/4"	1 1/4"	2"	1 1/4"	1 1/4"	1"	1 1/4"	.731"	1"
C	1'-1"	1 1/4"	2 1/4"	2"	1 1/4"	1 1/4"	1"	1 1/4"	.731"	1"
C	1'-1"	1 1/4"	2"	2 1/4"	1 1/4"	1 1/4"	1"	1 1/4"	.731"	1"
D	1 1/4"-1 1/2"	1 1/2"	2 1/4"	2 1/4"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	1.10"	1 1/4"
E	1 1/2"-1 3/4"	2 1/4"	2 1/4"	3"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	1.4"	1 1/4"
E	1 3/4"-1"	2 1/4"	2 1/4"	3"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	1.4"	1 1/4"
F	1"-1 1/4"	2 1/4"	1 1/4"	3 1/4"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	1.6"	1 1/4"
H	1 1/4"-1 1/2"	3 1/4"	1 1/4"	3 1/4"	2 1/4"	2 1/4"	2 1/4"	1.96"	1 1/4"	1 1/4"
I	1 1/2"-2"	3 1/4"	1 1/4"	3 1/4"	2 1/4"	2 1/4"	2 1/4"	2.42"	1 1/4"	1 1/4"
I	1 1/2"-2 1/4"	3 1/4"	1 1/4"	3 1/4"	2 1/4"	2 1/4"	2 1/4"	2.6"	1 1/4"	1 1/4"
J	1 1/4"-2 1/4"	4"	2 1/4"	4"	3 1/4"	2 1/4"	3 1/4"	3.1"	1 1/4"	1 1/4"

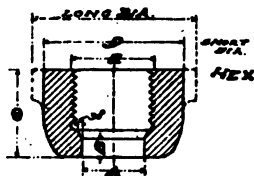
PUNCHES.



No. of Coupling.	Diameter of Punch at "X."							A.	B.	C.	D.	E.	Angle.
B	1"	1 1/4"	1 1/2"	1"	1 1/4"	1 1/2"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	1"	1 1/4"	30
C	1"	1 1/4"	1 1/2"	1"	1 1/4"	1 1/2"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	1"	1 1/4"	"
D	1 1/4"	1 1/2"	1 3/4"	1 1/4"	1 1/2"	1 3/4"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	"
E	1 1/2"	1 3/4"	2"	1 1/2"	1 3/4"	2"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	"
F	1 1/4"	1 1/2"	1 3/4"	1 1/4"	1 1/2"	1 3/4"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	"
G	1 1/4"	1 1/2"	1 3/4"	1 1/4"	1 1/2"	1 3/4"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	"
H	1 1/4"	1 1/2"	1 3/4"	1 1/4"	1 1/2"	1 3/4"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	"
I	1 1/4"	1 1/2"	1 3/4"	1 1/4"	1 1/2"	1 3/4"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	1 1/4"	"
J	2 1/4"	2 1/2"	2 3/4"	2 1/4"	2 1/2"	2 3/4"	2 1/4"	2 1/4"	2 1/4"	2 1/4"	2 1/4"	2 1/4"	"
K	2 1/4"	2 1/2"	2 3/4"	2 1/4"	2 1/2"	2 3/4"	2 1/4"	2 1/4"	2 1/4"	2 1/4"	2 1/4"	2 1/4"	"
L	3 1/4"	3 1/2"	3 3/4"	3 1/4"	3 1/2"	3 3/4"	3 1/4"	3 1/4"	3 1/4"	3 1/4"	3 1/4"	3 1/4"	"

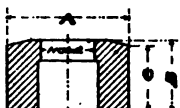
FIG. 126.

COUPLING NUTS



No. of Coupling.	A.	B.	C.	D.	E.	No. of Threads Per In.	No. of Coupling.	A.	B.	C.	D.	E.	No. of Threads Per In.
B	$\frac{1}{8}$ "	$\frac{1}{4}$ "	1"	$1\frac{1}{4}$ "	$1\frac{1}{2}$ "	12	H	$1\frac{1}{8}$ "	$\frac{3}{4}$ "	2"	$3\frac{1}{2}$ "	$2\frac{1}{2}$ "	12
C	$\frac{1}{8}$ "	$\frac{1}{4}$ "	$1\frac{1}{4}$ "	$1\frac{1}{4}$ "	1"	12	I	$2\frac{1}{8}$ "	$\frac{3}{4}$ "	$2\frac{1}{4}$ "	$3\frac{1}{2}$ "	$2\frac{1}{2}$ "	12
D	$\frac{1}{8}$ "	$\frac{1}{4}$ "	$1\frac{1}{4}$ "	$1\frac{1}{4}$ "	$1\frac{1}{4}$ "	12	J	$2\frac{1}{8}$ "	$\frac{3}{4}$ "	$2\frac{1}{4}$ "	$4\frac{1}{4}$ "	$3\frac{1}{4}$ "	12
E	$1\frac{1}{8}$ "	$\frac{1}{2}$ "	$1\frac{1}{4}$ "	$2\frac{1}{4}$ "	$1\frac{1}{4}$ "	12	K	$3\frac{1}{8}$ "	$1\frac{1}{4}$ "	$2\frac{1}{4}$ "	6"	$3\frac{1}{4}$ "	10
F	$1\frac{1}{4}$ "	$\frac{1}{2}$ "	$1\frac{1}{4}$ "	$2\frac{1}{4}$ "	$1\frac{1}{8}$ "	12	L	$4\frac{1}{8}$ "	$1\frac{1}{2}$ "	3"	$6\frac{1}{4}$ "	$4\frac{1}{2}$ "	10
G	$1\frac{1}{4}$ "	1"	$1\frac{1}{4}$ "	$2\frac{1}{4}$ "	2"	12							

DIES.



Diameter of Punch.	Hole in Die.	A.	B.	C.	Diameter of Punch.	Hole in Die.	A.	B.	C.
$\frac{1}{8}$ " - .125"	$\frac{1}{8}$ "	$1\frac{1}{8}$ "	1"	$1\frac{1}{4}$ "	$\frac{1}{8}$ " - .25"	$\frac{1}{8}$ "			
$\frac{1}{8}$ " - .156"	$\frac{1}{8}$ "				$\frac{1}{8}$ " - .28"	$\frac{1}{8}$ "			
$\frac{1}{8}$ " - .187"	$\frac{1}{8}$ "				$\frac{1}{8}$ " - .31"	$\frac{1}{8}$ "			
$\frac{1}{8}$ " - .218"	$\frac{1}{8}$ "				$\frac{1}{8}$ " - .34"	$\frac{1}{8}$ "			
$\frac{1}{4}$ " - .37"	$\frac{1}{4}$ "				$1\frac{1}{8}$ " - 1.62"	$1\frac{1}{8}$ "			
$\frac{1}{4}$ " - .40"	$\frac{1}{4}$ "				$1\frac{1}{8}$ " - 1.68"	$1\frac{1}{8}$ "			
$\frac{1}{4}$ " - .43"	$\frac{1}{4}$ "				$1\frac{1}{8}$ " - 1.78"	$1\frac{1}{8}$ "	$3\frac{1}{8}$ "	$1\frac{1}{4}$ "	$1\frac{1}{8}$ "
$\frac{1}{4}$ " - .46"	$\frac{1}{4}$ "				$1\frac{1}{8}$ " - 1.81"	$1\frac{1}{8}$ "			
$\frac{1}{4}$ " - .5"	$\frac{1}{4}$ "	$1\frac{1}{8}$ "	$1\frac{1}{4}$ "	1"	$1\frac{1}{8}$ " - 1.87"	$1\frac{1}{8}$ "			
$\frac{1}{4}$ " - .53"	$\frac{1}{4}$ "				$1\frac{1}{8}$ " - 1.94"	$2\frac{1}{8}$ "			
$\frac{1}{4}$ " - .56"	$\frac{1}{4}$ "				2" - 2"	$2\frac{1}{8}$ "	$4\frac{1}{8}$ "	$2\frac{1}{4}$ "	$2\frac{1}{8}$ "
$\frac{1}{4}$ " - .59"	$\frac{1}{4}$ "				$2\frac{1}{8}$ " - 2.06"	$2\frac{1}{8}$ "			
$\frac{1}{4}$ " - .62"	$\frac{1}{4}$ "				$2\frac{1}{8}$ " - 2.125"	$2\frac{1}{8}$ "			
$\frac{1}{4}$ " - .68"	$\frac{1}{4}$ "				$2\frac{1}{8}$ " - 2.187"	$2\frac{1}{8}$ "			
$\frac{1}{4}$ " - .75"	$\frac{1}{4}$ "	$2\frac{1}{8}$ "	$1\frac{1}{4}$ "	$1\frac{1}{8}$ "	$2\frac{1}{8}$ " - 2.25"	$2\frac{1}{8}$ "	$4\frac{1}{8}$ "	$2\frac{1}{4}$ "	$2\frac{1}{8}$ "
$\frac{1}{4}$ " - .81"	$\frac{1}{4}$ "				$2\frac{1}{8}$ " - 2.37"	$2\frac{1}{8}$ "			
$\frac{1}{4}$ " - .87"	$\frac{1}{4}$ "				$2\frac{1}{8}$ " - 2.5"	$2\frac{1}{8}$ "			
$\frac{1}{4}$ " - .93"	$\frac{1}{4}$ "	$2\frac{1}{8}$ "	$1\frac{1}{4}$ "	$1\frac{1}{8}$ "	$2\frac{1}{8}$ " - 2.63"	$2\frac{1}{8}$ "	$6\frac{1}{8}$ "	$2\frac{1}{4}$ "	$2\frac{1}{8}$ "
1" - 1"	$1\frac{1}{8}$ "				$2\frac{1}{8}$ " - 2.76"	$2\frac{1}{8}$ "			
$1\frac{1}{8}$ " - 1.06"	$1\frac{1}{8}$ "				$2\frac{1}{8}$ " - 2.87"	$2\frac{1}{8}$ "			
$1\frac{1}{8}$ " - 1.125"	$1\frac{1}{8}$ "	$2\frac{1}{8}$ "	$1\frac{1}{4}$ "	$1\frac{1}{8}$ "	3" - 3"	$3\frac{1}{8}$ "	$6\frac{1}{8}$ "	$2\frac{1}{4}$ "	$2\frac{1}{8}$ "
$1\frac{1}{8}$ " - 1.187"	$1\frac{1}{8}$ "				$3\frac{1}{8}$ " - 3.85"	$3\frac{1}{8}$ "			
$1\frac{1}{8}$ " - 1.25"	$1\frac{1}{8}$ "				$3\frac{1}{8}$ " - 3.5"	$3\frac{1}{8}$ "	$6\frac{1}{8}$ "	$2\frac{1}{4}$ "	$2\frac{1}{8}$ "
$1\frac{1}{8}$ " - 1.31"	$1\frac{1}{8}$ "				$3\frac{1}{8}$ " - 3.75"	$3\frac{1}{8}$ "			
$1\frac{1}{8}$ " - 1.37"	$1\frac{1}{8}$ "				4" - 4"	$4\frac{1}{8}$ "	7"	3"	$2\frac{1}{8}$ "
$1\frac{1}{8}$ " - 1.43"	$1\frac{1}{8}$ "	$3\frac{1}{8}$ "	$1\frac{1}{4}$ "	$1\frac{1}{8}$ "	$4\frac{1}{8}$ " - 4.125"	$4\frac{1}{8}$ "			
$1\frac{1}{8}$ " - 1.5"	$1\frac{1}{8}$ "								

Other Types of Shearing and Punching Machines.

The smallest sizes of punching and shearing machines are operated by hand power or foot power, medium sized machines are operated almost exclusively by belt and the largest machines are operated by belt, steam, water or electricity as shown in Figs. 127, 128, 129 and 130 respectively. These designs show present practice and are added to enable the designer to become more familiar with the form of the parts and the make up of the machines in general.

It will be noticed that in the larger machines the frame is of such a size as to project below the floor, the weight being carried on legs or lugs cast on the side of the frame. It will also be noticed that arrangements are made at the top of the frame for the attachment of a crane to assist in handling the material.

Most single machines have the lower end of the ram so constructed that either punches or shear blades may be attached. This requires some little time in changing and adjusting the tools. Double machines avoid the necessity of such changes.

Machines such as are here represented require more work than should be expected of one assignment, they may, however, be assigned to two men. This is especially true of the double machines, in which case the frames may be worked up independently, and the driving mechanism, jointly. Electric motor sizes and capacities may be obtained from any standard catalog of electric machinery.



FIG. 127.



FIG. 128.



FIG. 129.

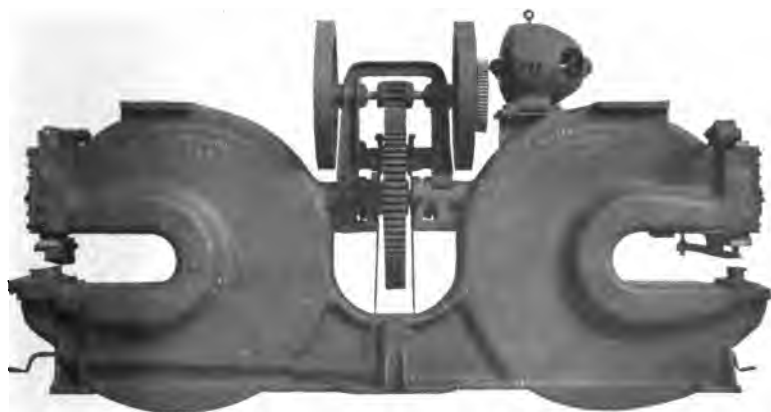
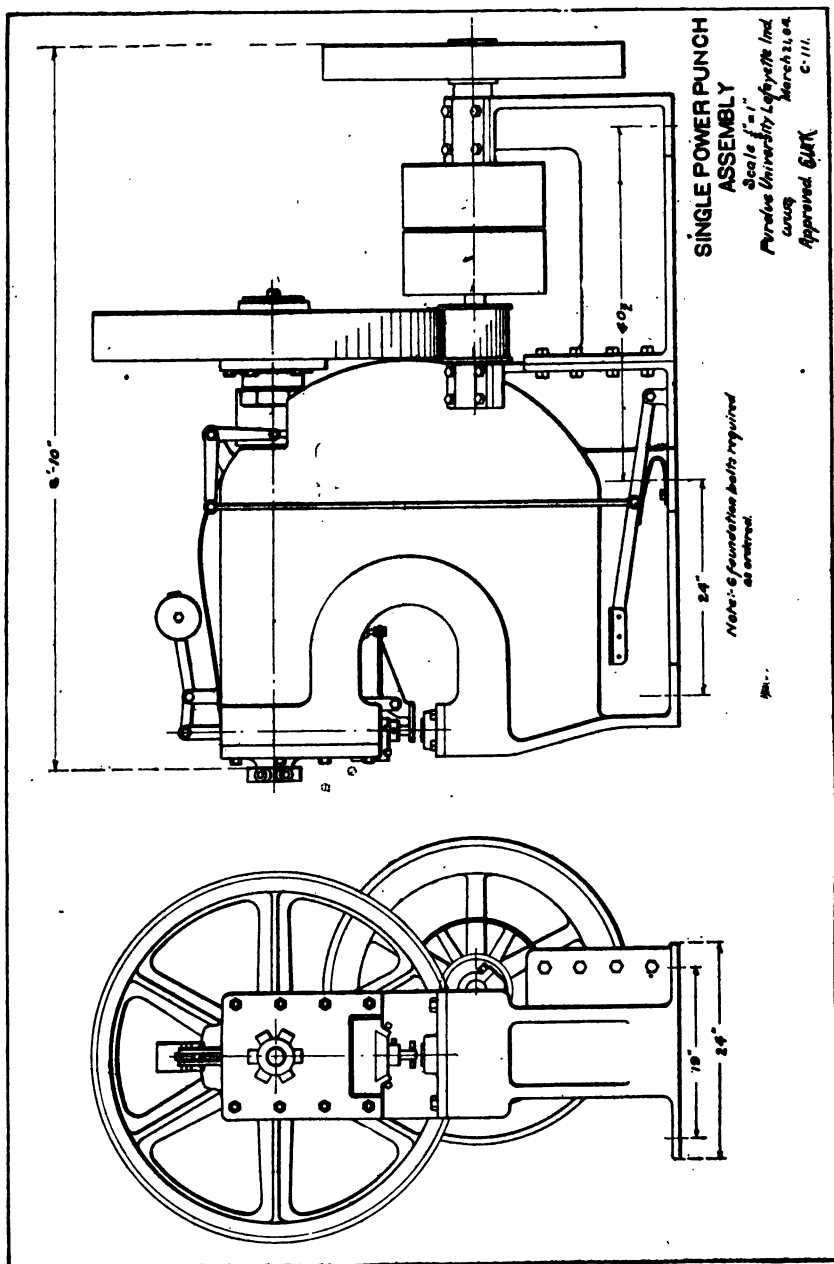
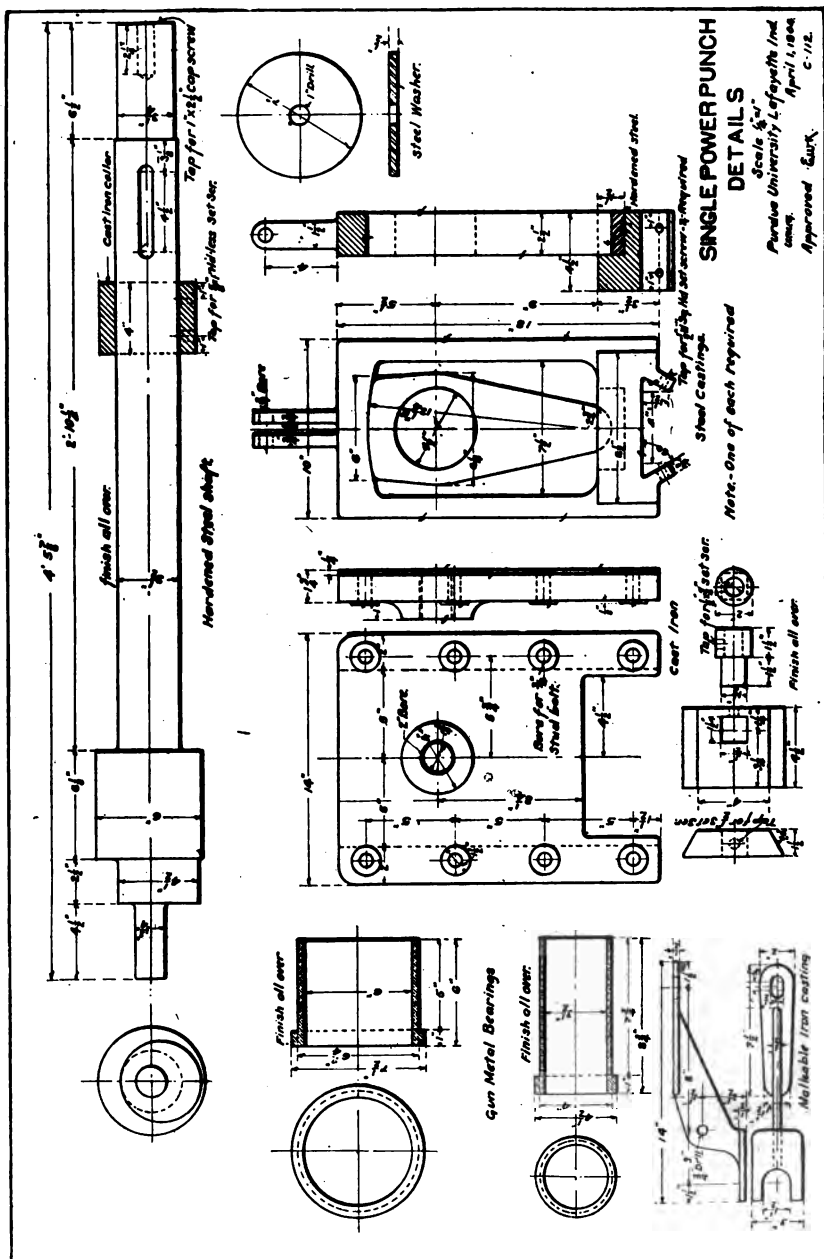
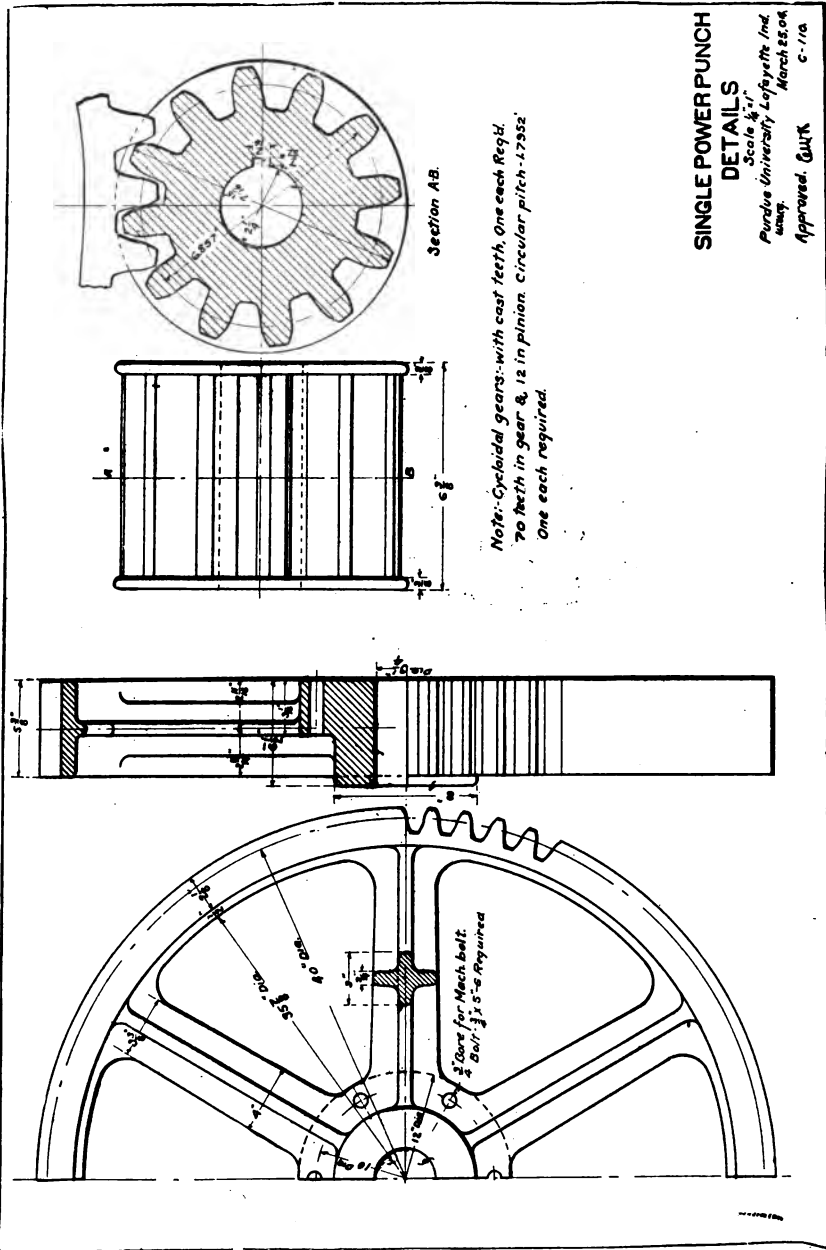
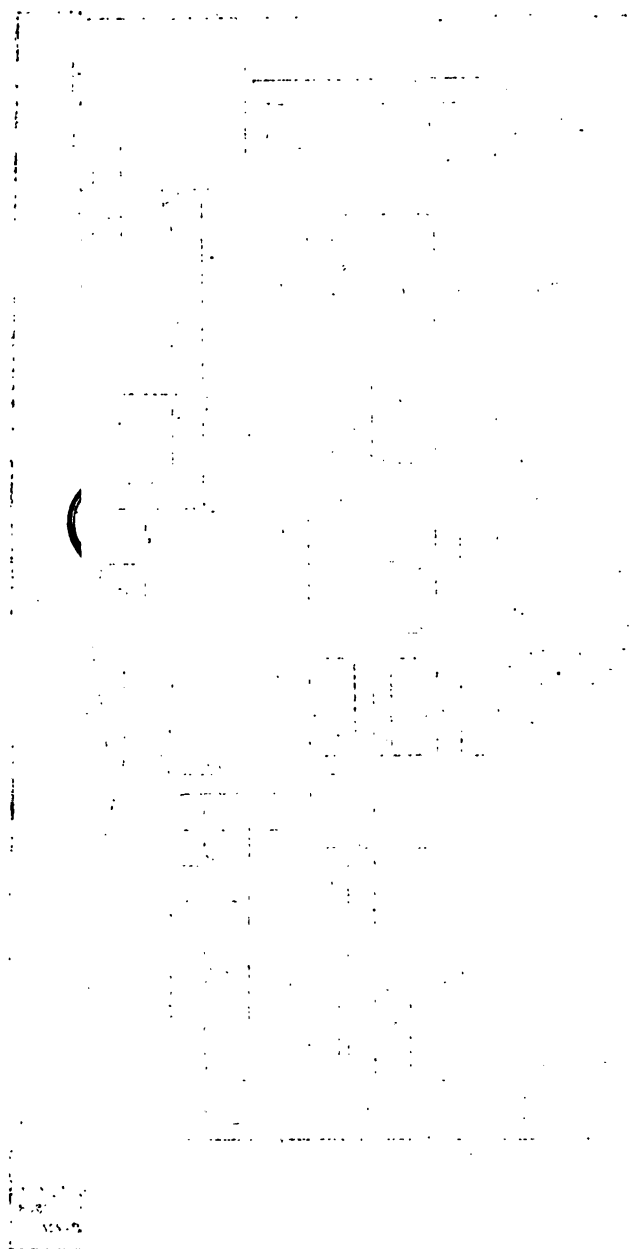


FIG. 130.

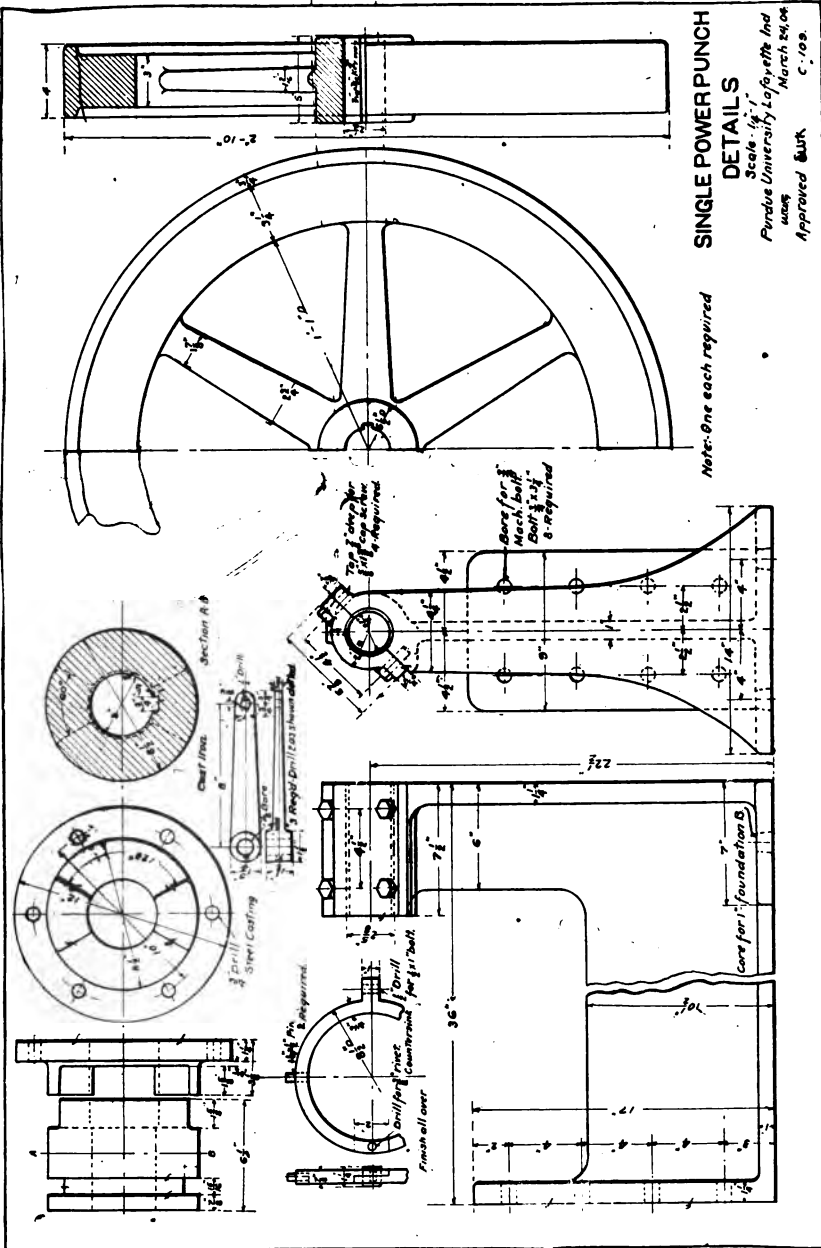








100-10



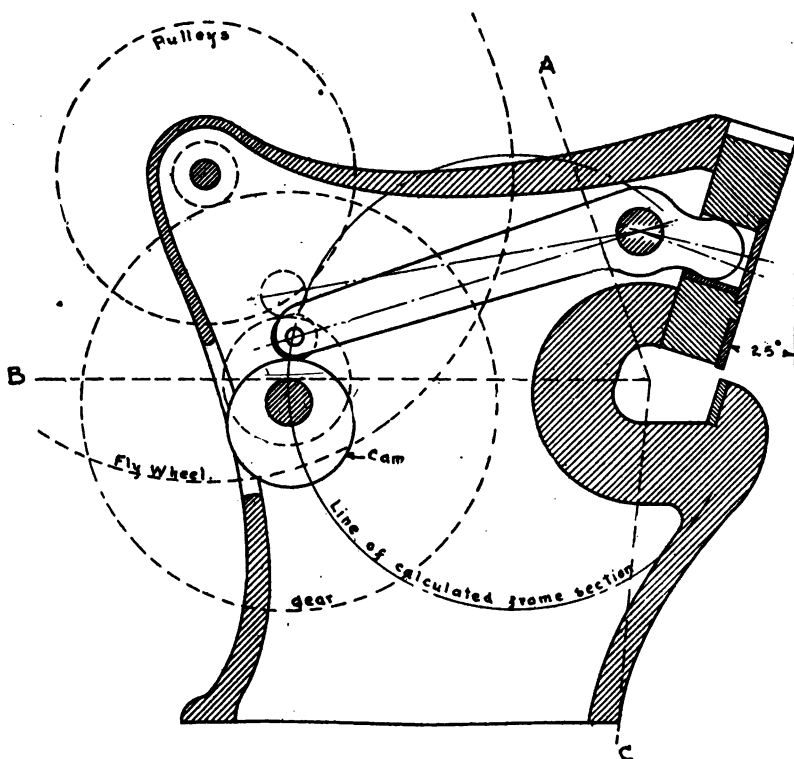
FIRST ALTERNATE, DESIGN NO. 2.

FIG. 131.

The Bevel Shear.

(Niles-Bement-Pond Catalog.)

207. Assignment:—

- Kind of material to be sheared.....
- Width of plate to be sheared, (6 to 12).....inches
- Thickness of plate to be sheared ($\frac{1}{4}$ to 1).....inches
- Depth of throat, (6 to 18).....inches
- Strokes of the ram per minute (15 to 20).....

The frame sections may be calculated if desired to a regular outline as shown in the dotted lines, after which modifications in this outline may be made by approximation. A better way, however, would be to sketch the approximate longitudinal frame section as above and figure for each of the several irregular sections, as *A*, *B* and *C*.

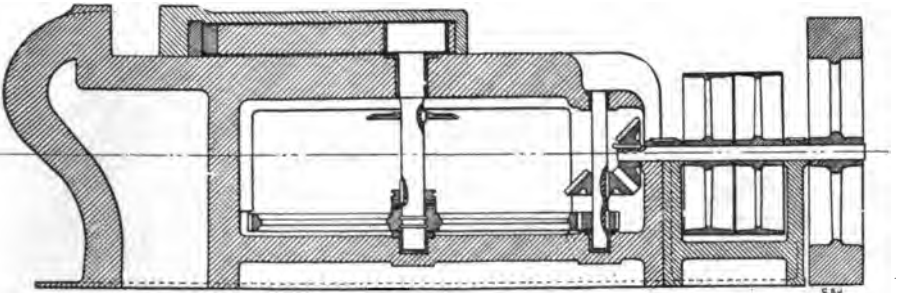
SECOND ALTERNATE, DESIGN NO. 2.

FIG. 132.

Horizontal Power Punch.

(Niles Tool Works Co., 1900 Catalog.)

208. Assignment:—

- Kind of material to be punched.....
 Size of largest hole punched.....inches.
 Thickness of the plate.....inches.
 Distance of centre of hole from edge of plate.....inches.
 Number of holes punched per minute.....

Horizontal punching machines may be designed in the same general way as the one described in the notes. It will be found that the frame sections may be calculated in the same way although the frame not being so regular will require a little more care in selecting the shapes and sizes of the various parts of the sections.

Machines of this type usually have a more shallow throat than the vertical type.

The line of the punch centre may be raised from the centre of the ram to the upper edge and is found convenient when punching near a shoulder.

THIRD ALTERNATE, DESIGN NO. 2.

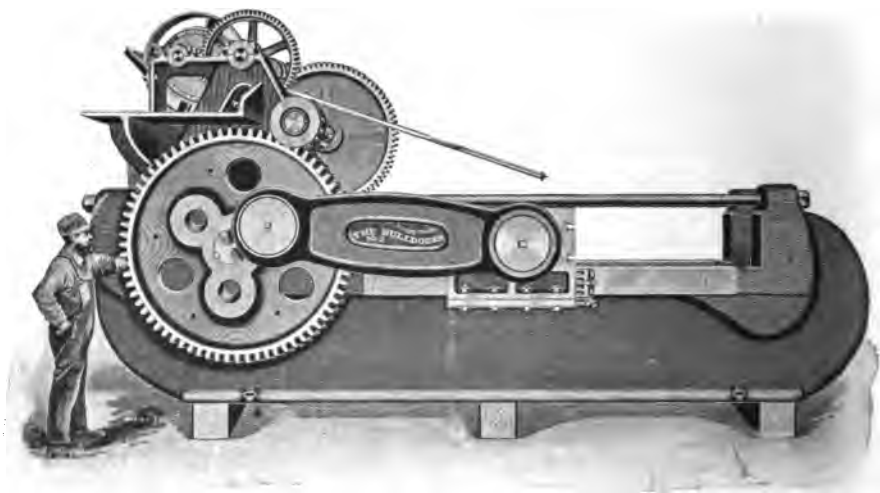


FIG. 133.

The Bulldozer.

209. Assignment.—

- Length of stroke (6 to 18).....inches.
- Maximum pressure (5000 to 30000).....pounds
- Number of strokes per minute (10 to 15).....

The Bulldozer, one of the most powerful of the horizontal presses, is used in forming or squeezing metals to shape between large dies in such processes as upsetting and bolt heading. It is also occasionally used in punching and straightening. The dies are very heavy, and sometimes the stroke is made long enough to permit a number of dies being inserted at one time, so as to allow several operations on the specimen without reheating.

Assume a typical work card, having the ordinates represent total pressures in pounds and the abscissas represent per cents of stroke. Let the work to be performed be such that the dies first strike the specimen at 25 per cent. of the stroke, also let it require the following total pressures to complete the work:

Per cent. of stroke.....	25	30	35	40	50	60	70	80	90	95	100
Per cent. of max. pressure..	00	60	70	75	80	82	78	75	75	80	100

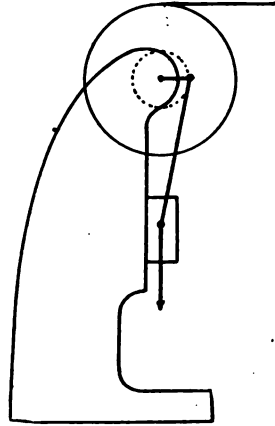
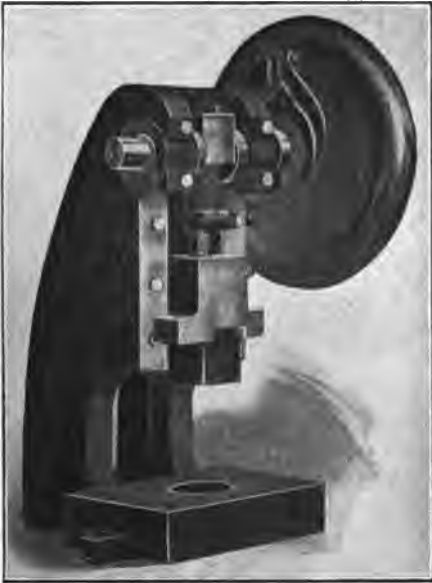
FOURTH ALTERNATE, DESIGN NO. 2.

FIG. 134.

Atlas Power Press.

(Atlas Machine Co., Catalog.)

The press shown in Fig. 134 is designed to take the place of the ordinary foot-press in doing light blanking, perforating, riveting, forming and closing. The clutch is of the standard Johnson type. A ball and socket joint between the shaft and the gate gives the latter a vertical adjustment of about $1\frac{1}{2}$ inches. The machine is furnished with a combination pulley and balance wheel. The mechanism of the machine is shown to the right. The following approximate sizes may be used for checking:

From bed to gate in lowest position.....	6 to 7 inches.
Stroke	$1\frac{1}{2}$ inches.
Distance between uprights.....	$3\frac{5}{8}$ to 6 inches.
Bed surface	7 x 10 to 8 x 12 inches.
Weight of wheel	50 to 100 pounds.

210. Assignment:—

P (crank 5 degrees from vertical, 1000 to 2000).....	pounds.
T (depth of gap) = (4 to 6).....	inches.
Revolutions per minute (200 to 300).....	

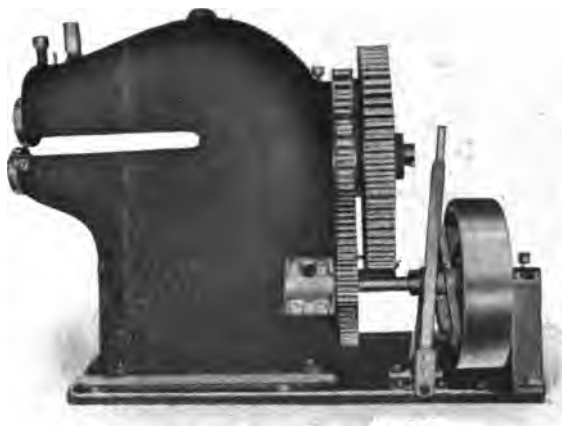
FIFTH ALTERNATE, DESIGN NO. 2.

FIG. 135.

The Lennox Rotary Shear.

(Joseph T. Ryerson, Catalog.)

(Bethlehem Foundry and Machine Co. Catalog.)

211. Assignment:—

Kind of material to be sheared.....
 Depth of throat (6 to 36).....inches
 Thickness of plate ($\frac{1}{8}$ to $\frac{3}{4}$).....inches.
 Diameter of cutters (6 to 12).....inches.
 Rim velocity of cutters (600 to 1000 feet per hour).....

Notation.

Shaft *L* is adjustable at lower end by screw *H* around *R* as a pivot. (See Fig. 137.)

Shaft *N* is adjustable in line parallel to center of shaft by nut *T*. *J* and *O* are gears of same diameter. The main pulley of the machine runs from 150 to 250 revolutions per minute. The cutters at *D* may be set apart a distance as great as one-fourth the thickness of the plate; the exact amount can best be determined by the experience of the operator. The exact force *W* at the cutters tending to rupture the frame is rather an indeterminate quantity but a safe value may be found by the following formula:

$$W = Af = tf \sqrt{tr - \frac{t^2}{4}} \left[1 - \left(.6834 + \frac{t^2}{40r} \right) \right]$$

where *t* = thickness of the metal to be cut; *r* = radius of the cutters; *f* = the ultimate shearing strength of the metal; and *A* = area of the metal being cut at any time, assuming the cutters to be in contact at the center line.



FIG. 136.

Lennox Rotary Bevel Shear.

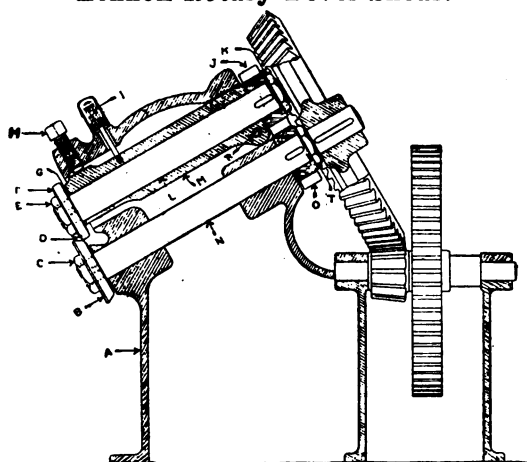


FIG. 137.

Section of Shear.

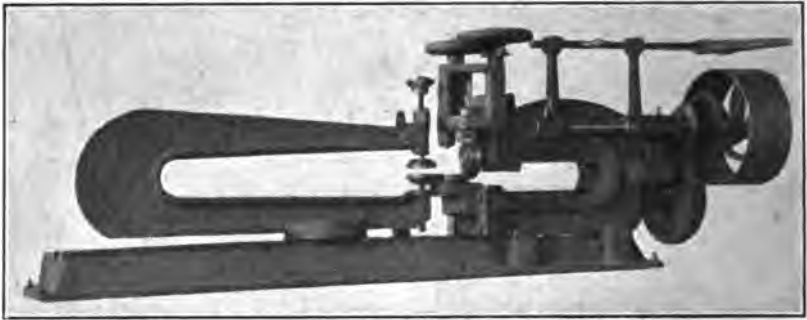
SIXTH ALTERNATE, DESIGN NO. 2.

FIG. 138.

Sheet Metal Flanger and Disk Cutter.
(Niagara Machine and Tool Works Catalog.)

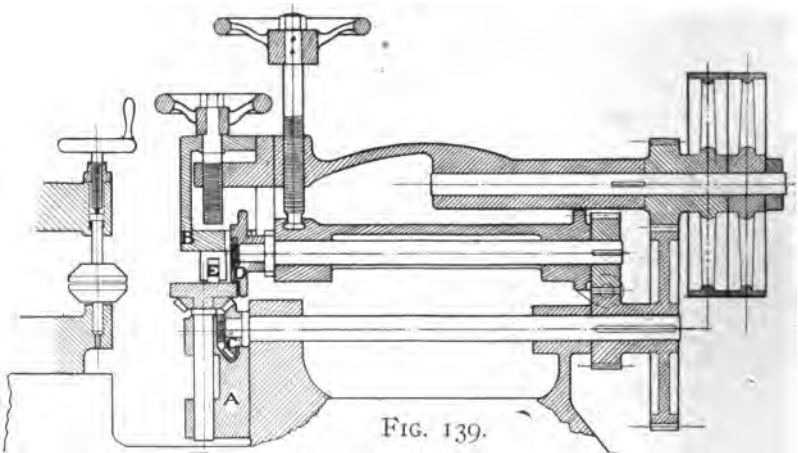
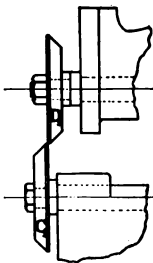


FIG. 139.



The section shows the machine with flanging rolls. These may be changed to cutter rolls as shown at the left. Other small rollers hold the metal to the plate while being operated upon. Size of flanges obtained in soft sheet steel as follows: 10 to 16 gage, $\frac{5}{8}$ to 1 inch; 16 to 20 gage, $\frac{3}{8}$ to $\frac{5}{8}$ inch; 22 to 24 gage, $\frac{1}{2}$ inch. Machine will cut up to No. 8 gage and flange to No. 10 gage.

212. Assignment:—

P = (1000 to 2000).....pounds.
 T (Depth of throat on machine, 12 to 20).....inches.
 G (Depth of throat on circle arm, 30 to 40).....inches.
 Speed of the tool (15 to 20).....revolutions per minute.

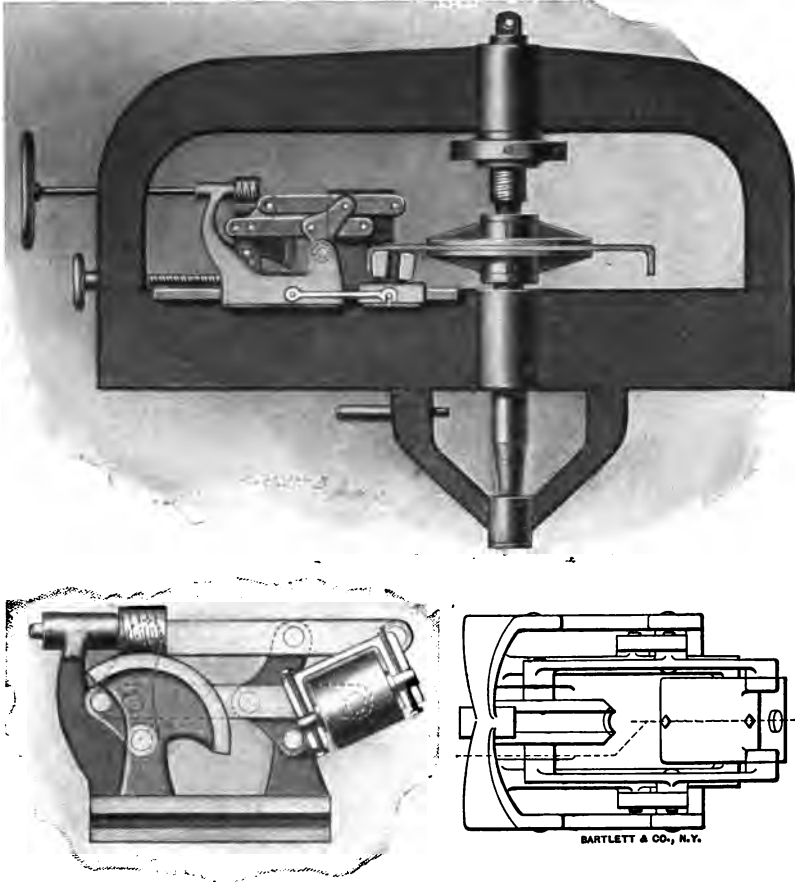
SEVENTH ALTERNATE, DESIGN NO. 2.

FIG. 140.

Boiler Head Flanging Machine Details.
 (Niles-Bement-Pond Catalog.)

213. Assignment.—This problem consists essentially of the development of the mechanism and the design of the parts shown in the detailed figures, i. e., the *flanging mechanism*. For application of these parts to the machine, see catalog. Develop the mechanism so that the rollers will revolve about each other with a uniform clearance in all positions. Assume a maximum thrust at the roller of (10,000 to 100,000) pounds, and design the parts so they will be sufficiently rigid to protect from flexure and breaking, and so the pull on the hand wheel will be within the capacity of one man, say 150 pounds.

CHAPTER VII.
DESIGN NO 3.

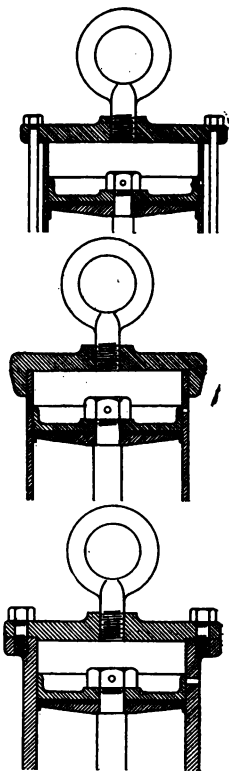
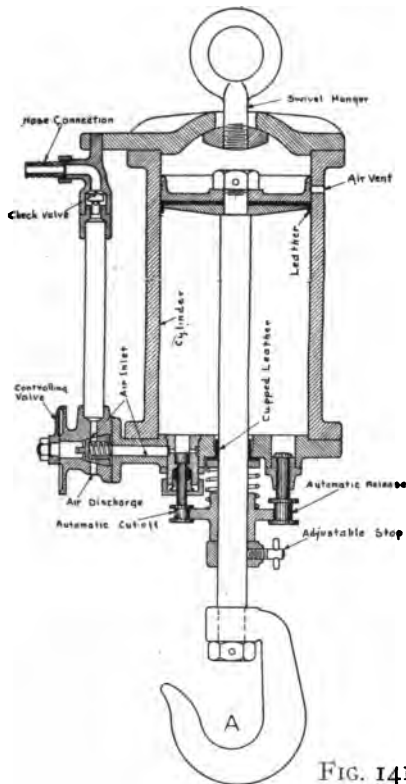


FIG. 141.

The Air Hoist. (Whiting Foundry Equipment Co.)

214. Assignment:—

- Capacity in free load.....pounds
- Weight of parts and friction.....per cent.
- Air pressure (80 to 100).....lbs. per sq. inch.
- Lift (2 to 4).....feet.

The following data may be used for checking, air at 80 pounds per square inch.

Diam. of Hoist	Size of pipe	Air consumed per 4 foot lift.
3 inches	1/2 inch	1.17 cu. ft.
7 "	3/4 "	6.63 " "
10 "	1 "	13.50 " "
16 "	1 "	34.49 " "

FIRST ALTERNATE, DESIGN NO. 3.**The Allen Riveters.**

(Joseph T. Ryerson and Son. Catalog.)

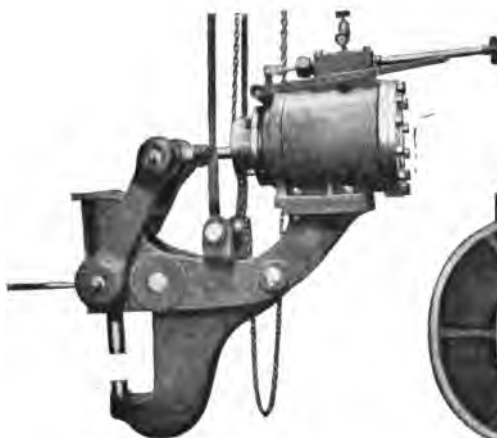


FIG. 142.
Lattice Column Type.



FIG. 143.
Jaw Type.

In this type of machine the piston rods connect levers of different lengths, thus forming a toggle joint. It very properly embodies features of both designs, No. 1 and No. 2. It may be assigned as an advanced substitute for No. 1 or as an extra.

The following maximum pressure necessary to set rivets may be expected in average practice.

Diameter of rivets in inches.....	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$
Pressure in tons of 2000 pounds.	25	30	40	50	65	80	100	150

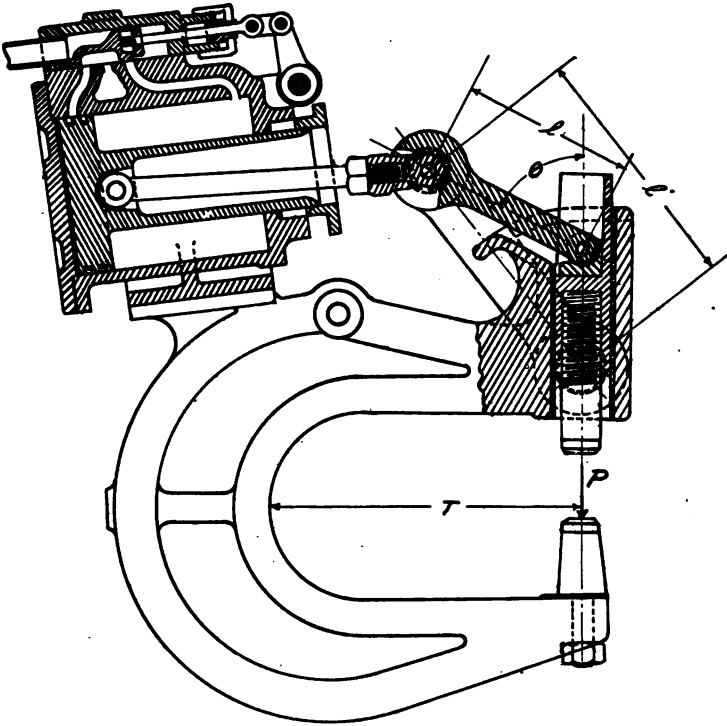


FIG. 144.

215. Assignment:—

$P = (8000 \text{ to } 50000) \dots\dots\dots \text{pounds.}$

$p = (\text{Air pressure } 80 \text{ to } 100) \dots\dots\dots \text{lbs. per sq. in.}$

$T = (4 \text{ to } 12) \text{ L. C. Type.} \dots\dots\dots \text{inches.}$

$T = (16 \text{ to } 66) \text{ J. Type.} \dots\dots\dots \text{inches.}$

$l = \dots\dots\dots \text{inches.}$

$l' = \dots\dots\dots \text{inches.}$

$\theta \text{ (min.)} = \dots\dots\dots \text{degrees.}$

Second Alternate, Design No. 3.

FIG. 145.

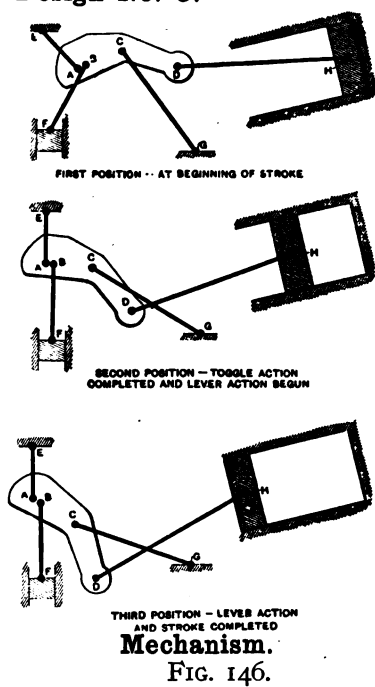


FIG. 146.

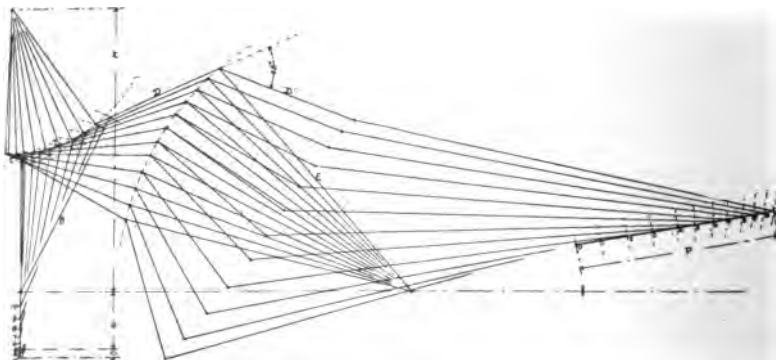
The Hanna Riveter.

FIG. 147.

Development of the Mechanism.

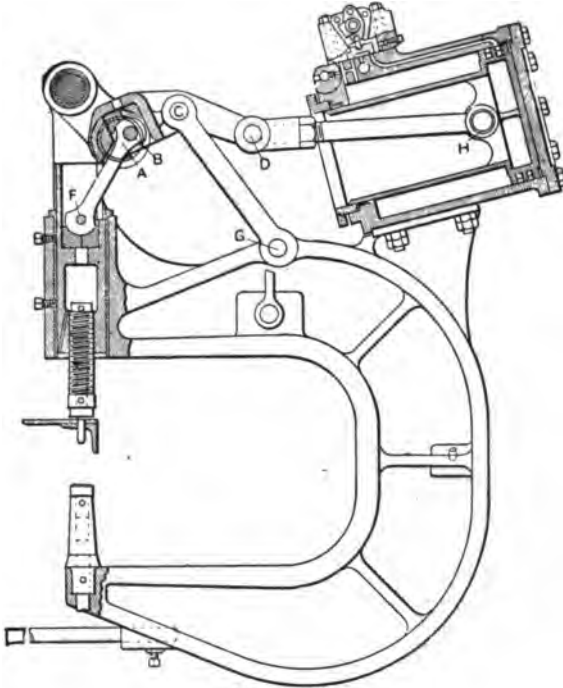


FIG. 148.

216. Assignment:—

$P = (50000 \text{ to } 200000) \dots\dots\dots$ pounds.

$T = (10\frac{1}{2} \text{ to } 66) \dots\dots\dots$ inches.

Maximum movement of the die ($O + N$) inches.

Air pressure.....pounds per square inch.

Assign Plunger Travel, $O + N$

Approximate other dimensions to table

A	B	C	D	D'	E	F	M	N	O	P
9	12	1	$7\frac{1}{2}$	9	18	26	$8\frac{3}{4}$	$3\frac{1}{2}$	$\frac{1}{2}$	12

First calculate and obtain the sizes for the frame, then give lengths and locate levers E A, B F, C G and D H such that the first half of the piston movement will cause a constantly decreasing velocity of the die, and the last half of the piston movement will cause a uniform velocity of the die. As an illustration of the above, in one machine the piston movement was 12 inches, the total movement of the die was 4 inches, the first 5 inches of piston travel gave a constantly decreasing velocity of the die through $3\frac{1}{2}$ inches of the die movement, leaving the last 7 inches of piston movement to produce a uniform velocity of the die through the last half inch of die movement.

THIRD ALTERNATE, DESIGN NO. 3.

FIG. 149.

The Alligator Riveter.

(Jos. T. Ryerson and Son. Catalog.)

217. Assignment:—(See Notation on First Alt. No. 3.) $P = (25 \text{ to } 65) \dots \dots \dots \text{Tons.}$ p (air pressure, 80 to 100) $\dots \dots \dots \text{lbs. per sq. inch.}$ $T = (9 \text{ to } 14) \dots \dots \dots \text{inches.}$ θ (min.) = $\dots \dots \dots \text{degrees.}$ Maximum movement of the dies (2 to 4) $\dots \dots \dots \text{inches.}$

Assume the length of the arm of the scissors such that the force to be transmitted through the toggle will not be so great as to require too large a cylinder. Also observe that a long arm requires a long toggle link and hence a long piston movement. This style of machine is used largely in structural and car shops. It may be made vertical or horizontal type. The dies are adjustable. The height of the gap varies from 6 to 14 inches.

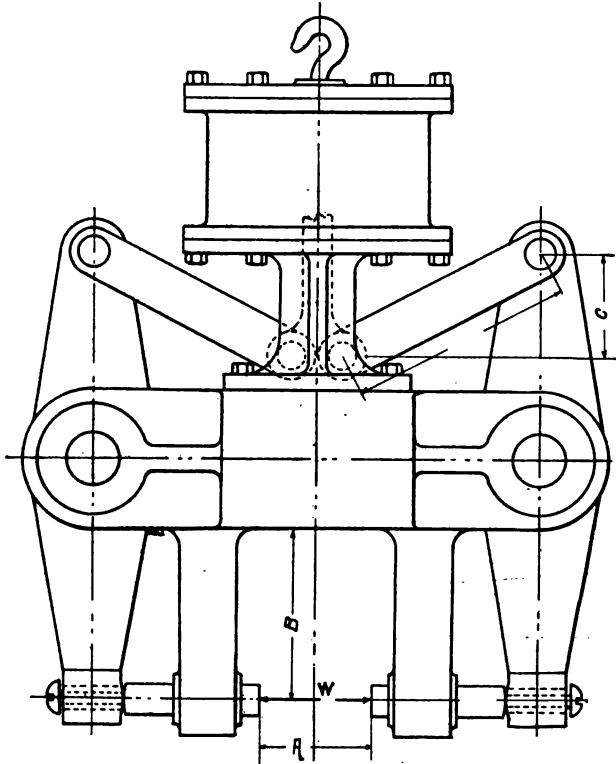
FOURTH ALTERNATE, DESIGN NO. 3.

FIG. 150.

Mudring Riveter.**218. Assignment:—**

$W = (25,000 \text{ to } 100,000)$ pounds.

p (air pressure, 80 to 100) pounds.

$T = B = (8 \text{ to } 16)$ inches.

$A = (5 \text{ to } 8)$ inches.

C (min.) when $\theta =$ degrees.

Total die movement = $(3 \text{ to } 5)$ inches.

This design may be modified by having the cylinder enclosed within the base if desired. In such an arrangement the piston rod becomes a compression member. Design also for air pipes and valves.

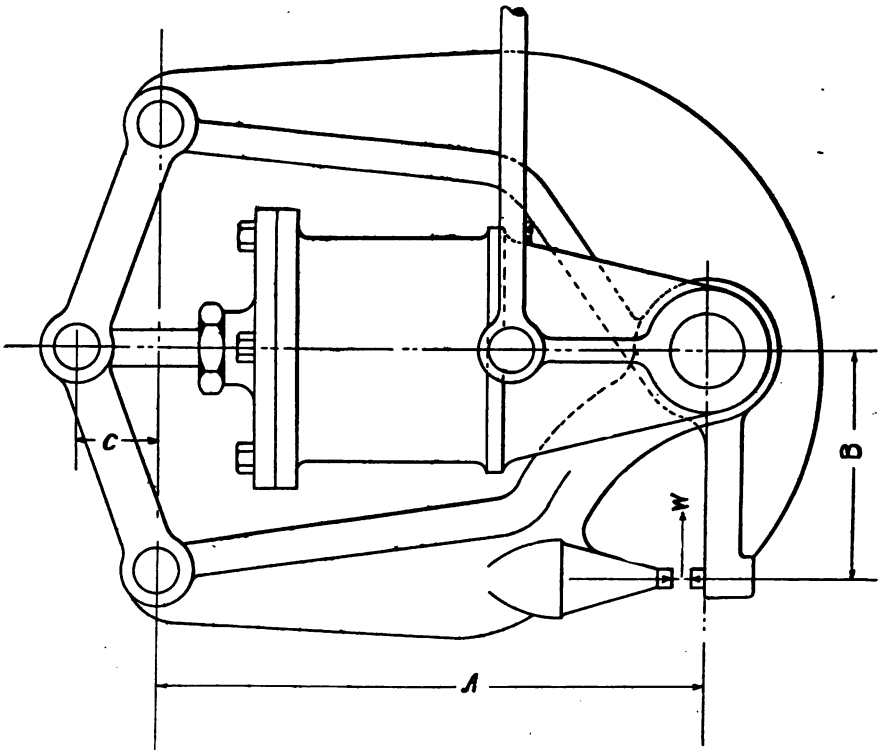
FIFTH ALTERNATE, DESIGN NO. 3.

FIG. 151.

Lever Riveter.**219. Assignment:—** $W = (20000 \text{ to } 60000) \dots \text{pounds.}$ p (air pressure 80 to 100) $\dots \text{pounds.}$ T (throat, 8 to 12) $\dots \text{inches.}$ $A = (20 \text{ to } 36) \dots \text{inches.}$ C (min.) when $\theta = \dots \text{degrees.}$ Total die movement (2 to 3) $\dots \text{inches.}$

When the arms are in their inner positions the cylinder must not touch them. Design also for air pipes and valves.

SIXTH ALTERNATE, DESIGN NO. 3.

FIG. 152.
25 Ton Portable.



FIG. 153.
50 Ton Portable.

Hydraulic Riveting Machine.
(Niles-Bement-Pond Catalog.)

220. Assignment:—

$P = (15 \text{ to } 50) \dots \dots \dots \text{tons.}$

$p = (800 - 1500) \dots \dots \dots \text{pounds per square inch.}$

$T = (6 - 15) \dots \dots \dots \text{inches.}$

Size of rivet (see First Alt. Des. No. 3)inches.

In this design the cylinder, frame, supports and valves are important in the order named. The piping is a feature that can be modified to suit almost any condition of frame. Such machines are used on structural and bridge work.

SEVENTH ALTERNATE, DESIGN NO. 3.
Triple Pressure Hydraulic Riveting Machine.
 (Niles-Bement-Pond Company.)



FIG. 154.

also be attached to main plunger so that intermediate sleeve *H* is locked to main plunger and moves with it. When so arranged the pressure on the dies is controlled by the difference between area of main plunger and intermediate sleeve *H*. Cover *F* over die slide *D* contains the push back piston *G* bearing directly upon main plunger.

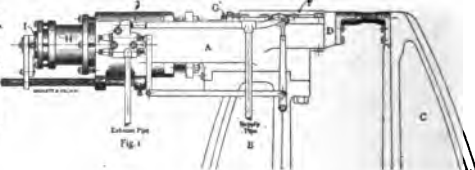


FIG. 155.

221. Assignment.—

$P = (50 \text{ to } 150) \dots \text{tons}$

$p = (1500 \text{ to } 2000) \dots \text{pounds per square inch.}$

$T = (12 \text{ to } 17) \dots \text{feet}$

This machine is built with three capacities: 50, 100 and 150 tons, for driving $\frac{7}{8}$, $1\frac{1}{4}$ and $1\frac{1}{2}$ inch rivets. The gap *T* is made in two lengths 12 and 17 feet. The cylinder is designed for three pressures of water, the highest being 1500 to 2000 pounds per square inch. By means of the three pressures provided as per section (see also catalog) the distributing value is not needed.

"On frame *B* is mounted cylinder *A* with main plunger *E*. To the main plunger can be attached, by means of the interrupted thread and nut *J*, the small plunger *I*. When so arranged plungers *E* and *I* move together and pressure on the dies is equivalent to the pressure *p* on the difference between the two areas. Small plunger can

CHAPTER VIII.

STUDIES IN THE KINEMATICS OF MACHINES.

The following problems in kinematics are given to supplement the work in both mechanism and design. One or more of these problems may be assigned between designs 1 and 2 and between 2 and 3 and will serve as a relaxation from the tedium of the longer problems of design. In their solution they contemplate pure mechanism (line motion) only and will not deal in any way with the strength or proportion of parts. The problems are arranged in a graduated series: First, those distinctly outlined and requiring little or no originality; second, those open to original ideas, but having one solution suggested, out of a number that might be made; third, those open to a number of solutions but requiring complete originality and invention.

A series of illustrative problems in the study of the mechanical movements of machines was given in the *American Machinist*, one problem each month, beginning December 1, 1904. In order that originality be developed, it is suggested that these problems be read in connection with the assignment given.

The kinematic problems relating to valve gears and link motions are classified at the last of the list and may be given between designs 2 and 3 or after design 3, so as to be taken in parallel with or after the subject of Engines and Boilers.

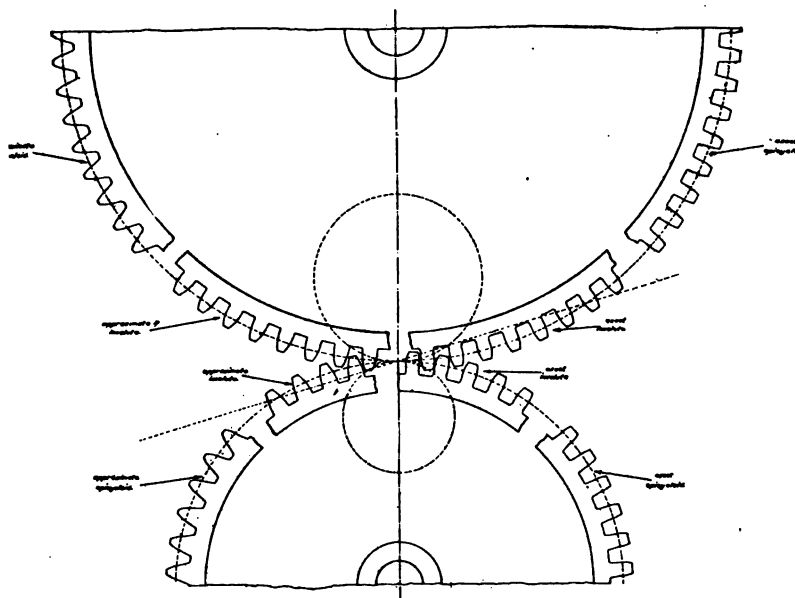
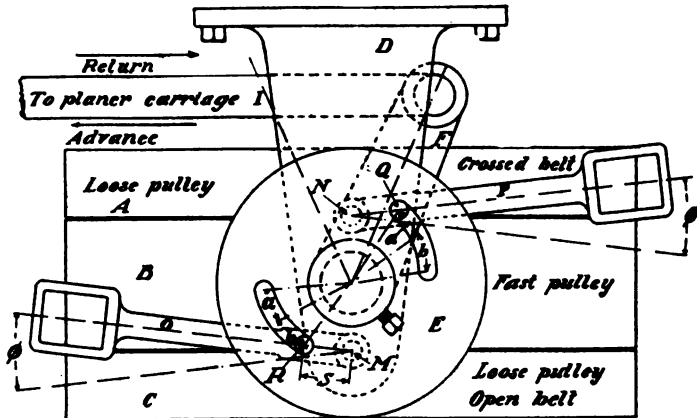
KINEMATIC SHEET NO. 1.

FIG. 156.

222. Assignment:—Given, a problem by which the diameters of the gears may be obtained. It is required to construct the tooth outlines for each by means of the following systems:

- Exact epicycloidal
- Exact involute
- Approximate epicycloidal
- Approximate involute.

KINEMATIC SHEET NO. 2.



a. Arc of circle.

b. Arc of cam.

FIG. 157.

Planer Cam.

223. Assignment:—

A and *C* are loose pulleys, *B* is a tight pulley.

D is fastened to frame of planer.

I moves back and forth, oscillating link *F* about *G*.

E is rigidly connected to *F* by set screw.

Levers *O* and *P* are pivoted at *m* and *n* on *D*.

Q and *R* are rollers fastened to the shifting levers.

Diameter pulleys (10-24)inches.

S =inches.

Width of fast pulley (3-10)inches.

Width of loose pulleys, each, (2-8).....inches.

Construct curve of cam so that the shifter will be constantly accelerated during first half of its motion and constantly retarded during latter half.

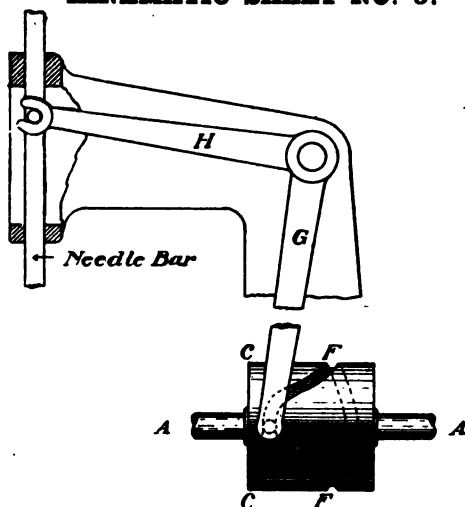
KINEMATIC SHEET NO. 3.

FIG. 158.

Cam of Home Sewing Machine.**224. Assignment:—**

Diameter of cylinder ($1\frac{1}{4}$ to $3\frac{1}{2}$).....inches.

Depth of Groove ($\frac{1}{8}$ to $\frac{1}{2}$).....inch.

Diameter of Roller ($\frac{3}{8}$ to $\frac{5}{8}$).....inch.

Stroke of bar (1 to 3).....inches.

Length of arm G (6 to 10).....inches.

Length of arm H (10 to 18).....inches.

Design a cylindrical cam similar to that shown in the sketch to engage a rocker arm. Divide the motion into 24 time periods. The follower is to move with a constant acceleration during four time periods; during the next eight periods it is to move uniformly with the velocity attained; during the next four periods it is to come to rest with a constant retardation. The return motion consists of eight time periods; during the first four periods it is to be constantly accelerated and during the remaining four periods it is to be constantly retarded.

Required full projection of cam outline on the cylinder. This will require the development of the cylinder at top and bottom of groove.

KINEMATIC SHEET NO. 4.

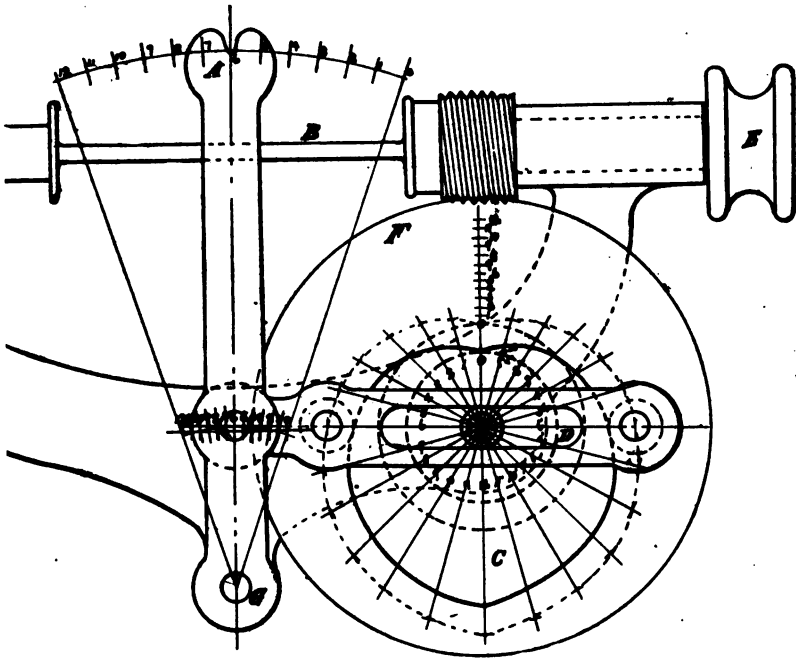


FIG. 159.

Sewing Machine Bobbin Winder

225. Assignment:—

Number of threads to be laid per inch of spool length (30 to 100)

Length of spool ($1\frac{1}{2}$ to $2\frac{1}{2}$).....inches.

KINEMATIC SHEET NO. 5.

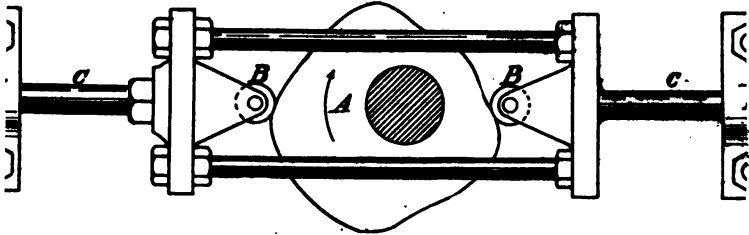


FIG. 160.

226. Assignment:—

Design the constant diameter cam, *A*, as shown, under the following conditions: Follower to move with harmonic motion from extreme right to left; to return one-half the distance by uniform motion; to remain at rest for one sixth the revolution of the cam, and to return to starting point by uniform motion. Total stroke of follower in one direction =inches.

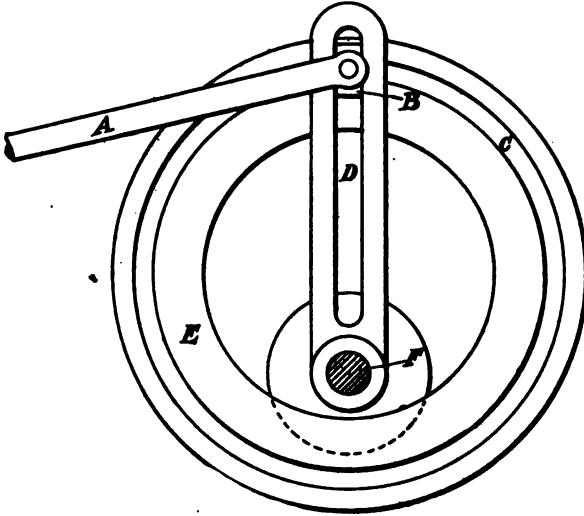
KINEMATIC SHEET NO. 6.

FIG. 161.

Quick Return Mechanism.**227. Assignment:—**

Length of lever, *A*, (18 to 24).....inches.

External diam. of circular slot (8 to 10).....inches.

Distance from center of rotating shaft, *F*, to center of circular slot, (4 to 10).....inches.

Plot velocity-time diagram of crosshead at end of arm *A*, which moves along a horizontal line through *F*.

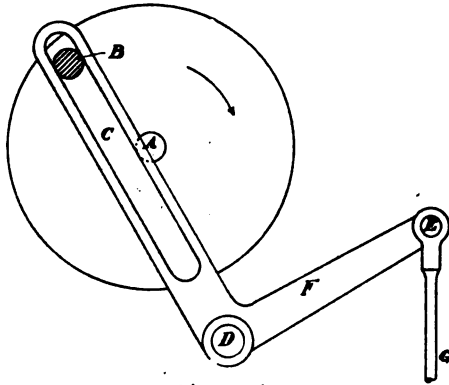
KINEMATIC SHEET NO. 7.

FIG. 162.

Quick Return Mechanism.**228. Assignment:—**

Radius of pin *B* from *A* (8 to 16).....inches.

Distance from *A* to *D*, (18 to 24).....inches.

Distance of *A* above horizontal line through *D*.....inches.

If *B* revolves with uniform rotation about *A*, plot the velocity-time diagram of block at lower end of *E G*.

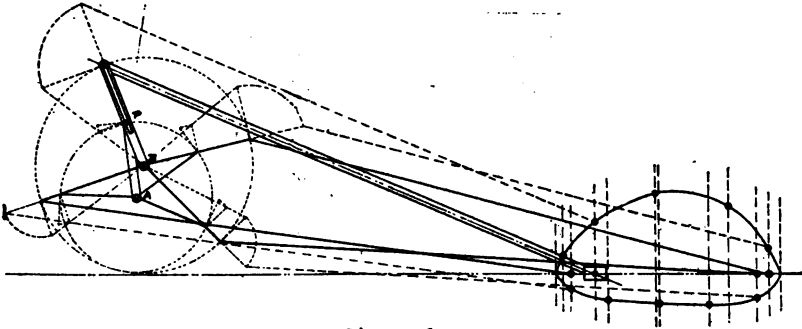
KINEMATIC SHEET NO. 8.

FIG. 163.

229. Assignment:—Lay out a Whitworth Quick Return motion, with the path of the tool below the centre *B* of the slotted crank *BP*, according to the following data:

Length of stroke.....inches

Length of connecting rod.....inches.

Length of A. P.....inches.

R. P. M. of crank.....

Period of advance to return of tool = 2 : 1

Construct the linear velocity—space diagram of the tool.

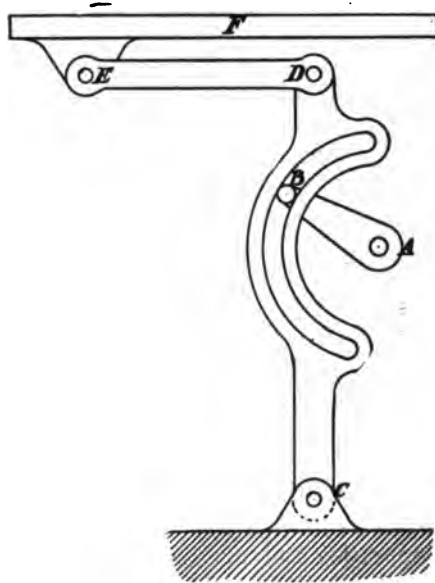
KINEMATIC SHEET NO. 9.

FIG. 164.

230. Assignment:—

Assume center A directly above center C , and that slot in which B works is on the arc of a circle, with radius AB . Plot velocity-time diagram for member F if AB rotates continuously, and members are proportioned as follows:

- Length AB (6 to 12).....inches.
 Length CD (18 to 30).....inches.
 Length DE (16 to 20).....inches.

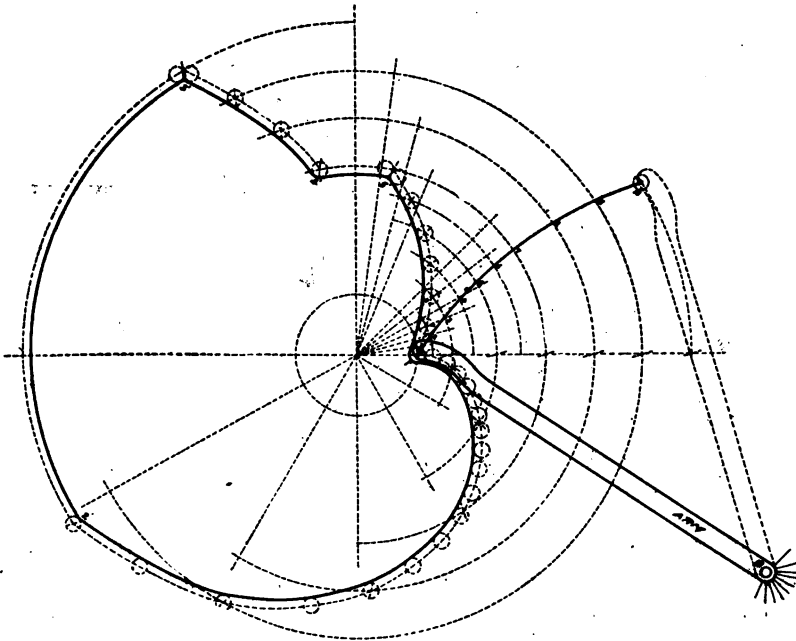
KINEMATIC SHEET NO. 10.

FIG. 165.

• **231. Assignment:**—Having given an oscillating arm, pivoted at point *B*, design a cam to move the end of the arm over the path 1, 2, 3, 4, 5.....13. The cam may have a uniform or varying motion while the arm may move uniformly or according to any law of motion desired.

KINEMATIC SHEET NO. 11.

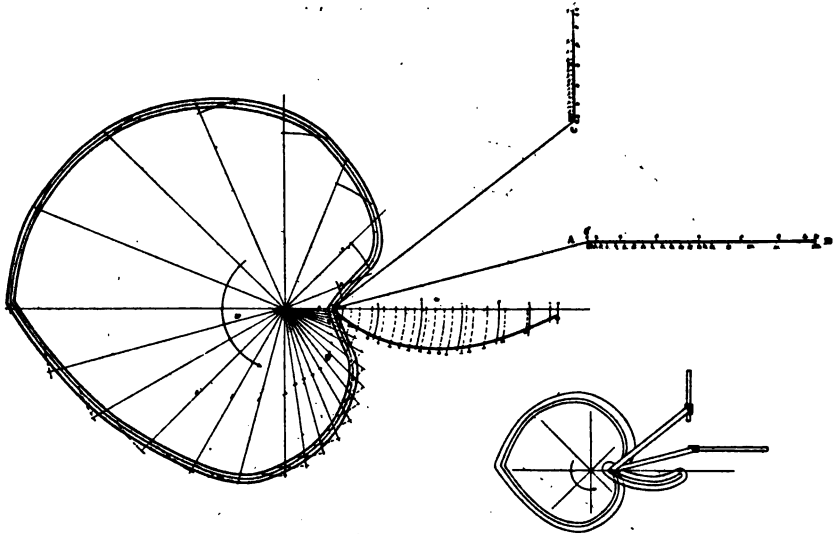


FIG. 166.

232. Assignment:—Two crossheads are to be driven in paths AB and CD intersecting at right angles. The length of the stroke, CD , is one-half that of AB . Motion is to be given to both crossheads by a single rotating cam. Such guides and connecting rods as are necessary may be employed. No part of the mechanism is to project within the angle DAB at any time. Motion away from A is to be according to the following schedule:

$\frac{1}{8}$ stroke, uniform acceleration.

$\frac{3}{8}$ stroke, uniform motion.

$\frac{1}{2}$ stroke, uniform acceleration.

Motion towards A to be harmonic.

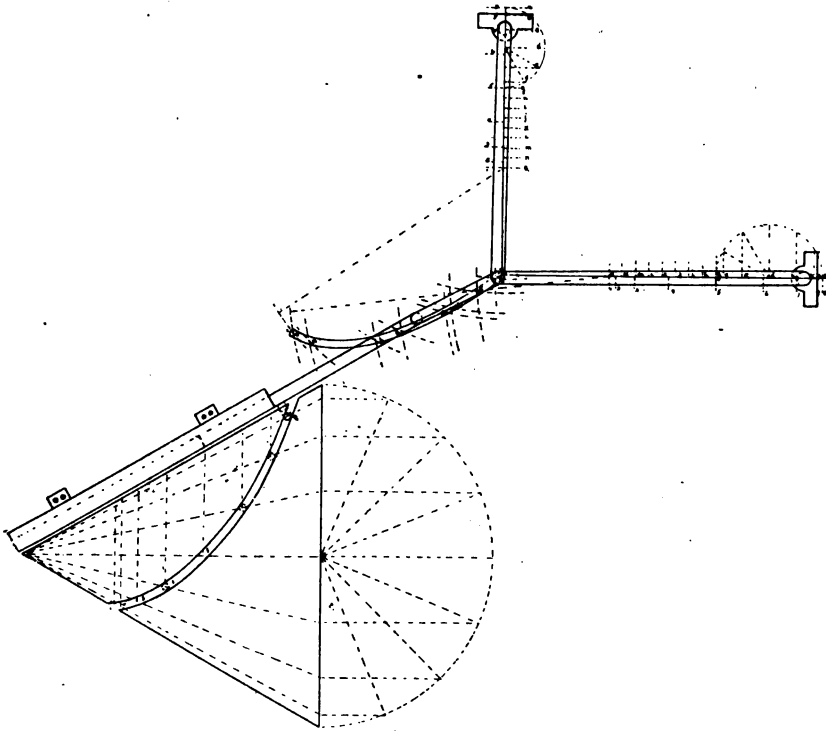
KINEMATIC SHEET NO. 12.

FIG. 167.

233. Assignment:—

Let vertical crosshead be *A*, horizontal crosshead be *B*, the pin connection be *C*, then *C* will travel through the stationary cam curve as shown.

Length of horizontal connecting rod.....inches.

Length of vertical connecting rod.....inches.

Travel of horizontal crosshead.....inches.

Travel of vertical crosshead.....inches.

Crossheads to move out inches with uniform acceleration; out inches with uniform motion; out inches with uniform acceleration; and to move in inches with increasing harmonic motion; in inches with uniform motion and in inches with decreasing harmonic motion.

Develop both top and bottom of groove in cone cam.

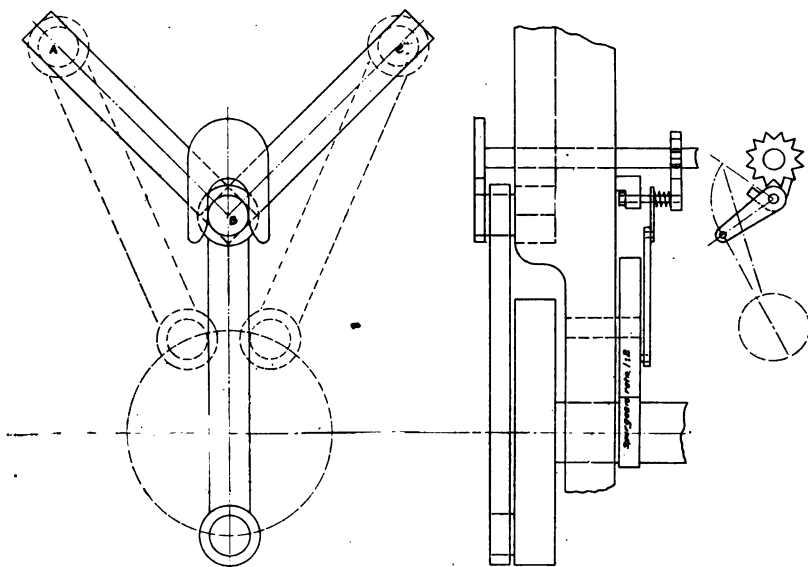
KINEMATIC SHEET NO. 13.

FIG. 168.

234. Assignment:—Having given the path of a groove ABC , a follower block is to move from A to B to C to B to A . Design a mechanism without the use of cams, and without allowing any part of the driving mechanism to extend within the angle ABC . Rack and pinion, or chain drives cannot be used directly to produce the motion.

KINEMATIC SHEET NO. 14.

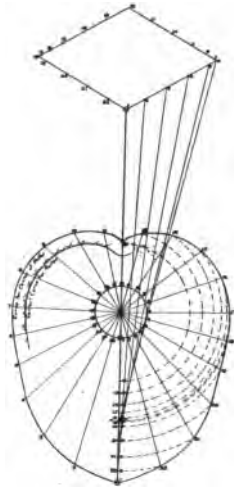
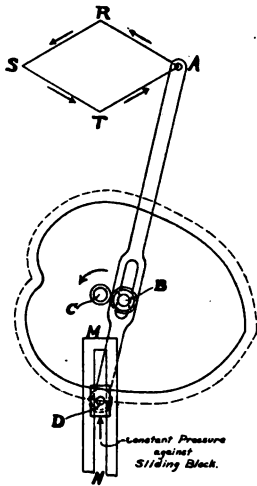


FIG. 169.

235. Assignment:—Having given any path $A R S T$ around which a point is to travel, design a mechanism to guide the point, the mechanism to have but one rotating shaft and one rotating disk cam, although other machine elements may enter into the construction. No part of the mechanism, excepting a single driving arm, shall project within the path $A R S T$, or above the horizontal line drawn through T .

The movement of the point will be

- A to R () of period of rotation.
 R to S () of period of rotation.
 S to T () of period of rotation.
 T to A () of period of rotation.



ORIGINAL KINEMATIC PROBLEMS

236. Assignment:—Sketches *A* to *E* show some of the common forms of paper clips on the market. The problem is to design cams, connecting levers and properly shaped dies to produce from a spool of wire some one of the forms indicated. Sketches may be taken as full size.

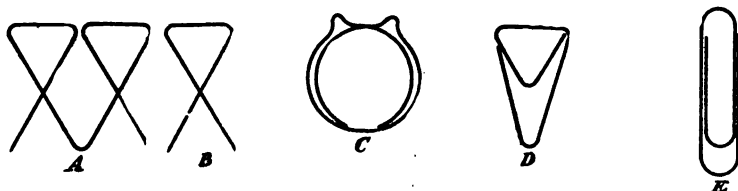


FIG. 170.

237. Assignment:—The path of a block consists of two parts, *AB* and *BC*. *BC* is $\frac{1}{2}$ the length of *AB* and perpendicular to *AB*

Motion cycle to be as follows:—

$\frac{1}{8}$ <i>B</i> to <i>A</i>	uniform acceleration.
$\frac{3}{4}$ <i>B</i> to <i>A</i>	uniform motion.
$\frac{1}{8}$ <i>B</i> to <i>A</i>	uniform acceleration.
<i>A</i> to <i>B</i>	harmonic motion.
$\frac{1}{8}$ <i>B</i> to <i>C</i>	uniform acceleration.
$\frac{3}{4}$ <i>B</i> to <i>C</i>	uniform motion.
$\frac{1}{8}$ <i>B</i> to <i>C</i>	uniform acceleration.
<i>C</i> to <i>B</i>	harmonic motion.

The motion of the block is to be obtained from a single disk cam, and no part of the mechanism—excepting a single guiding arm to impart motion to block—shall extend outside the angle *ABD*, where *D* is on a continuation of *CB*. Use not more than two levers or bell cranks and no connecting links, and have block make complete cycle in one revolution of cam.

238. Assignment:—The path of a block is to be a square *A, B, C, D*, the block to be driven by a single cylindrical cam rotating with a vertical shaft, i. e., shaft is perpendicular to plane of path. No part of the driving mechanism is to operate in the plane of the square. The motion cycle is to be:

$\frac{1}{8}$ <i>A</i> to <i>B</i>	uniform acceleration.
$\frac{3}{4}$ <i>A</i> to <i>B</i>	uniform motion.
$\frac{1}{8}$ <i>A</i> to <i>B</i>	uniform acceleration.

This to be repeated for *B* to *C*, *C* to *D*, and *D* to *A*.

239. Assignment:—A follower block has motion along a path $A B C D$. $A B$ and $D C$ are each perpendicular to $B C$, on the same side, and at the ends of, the line $B C$. In length, these path parts bear the following relations $B C = 2 A B = 1\frac{1}{4} D C$. One cylindrical cam is to be used, and no part of the driving mechanism is to extend within the figure $A B C D$, at any time during the motion, the cycle of which is to be:—

$\frac{1}{4} B$ to C	constant acceleration.
$\frac{1}{2} B$ to C	uniform motion.
$\frac{1}{4} B$ to C	constant deceleration.
$\frac{1}{8} C$ to D	constant acceleration.
$\frac{3}{4} C$ to D	uniform motion.
$\frac{1}{8} C$ to D	constant deceleration.
D to C	same variations as B to C .
C to B	same variations as B to C .
B to A , A to B	harmonic motion.

240. Assignment:—A follower block is to move in a groove whose center line is $A B C$. $B C$ is perpendicular to $A B$, and $\frac{3}{4}$ as long as $A B$. The motion is to be given by a single cylindrical cam, which may, however, carry more than one groove. Not more than two levers or bell cranks and not more than two connecting rods may be used. No part of the mechanism is to extend within the angle $A B C$, and the cam must lie in the angle made by prolonging $A B$ and $C B$.

$\frac{1}{8} A$ to B	constant acceleration.
$\frac{3}{4} A$ to B	constant motion.
$\frac{1}{8} A$ to B	constant deceleration.
$\frac{3}{8} B$ to C	increasing harmonic.
$\frac{1}{4} B$ to C	constant motion.
$\frac{3}{8} B$ to C	decreasing harmonic.
$\frac{1}{4} C$ to B	constant acceleration.
$\frac{1}{2} C$ to B	constant motion.
$\frac{1}{4} C$ to B	constant deceleration.
$\frac{1}{4} B$ to A	increasing harmonic.
$\frac{1}{2} B$ to A	constant motion.
$\frac{1}{4} B$ to A	decreasing harmonic.

MECHANISM OF THE RITES INERTIA GOVERNOR.

Rites Inertia Governor.

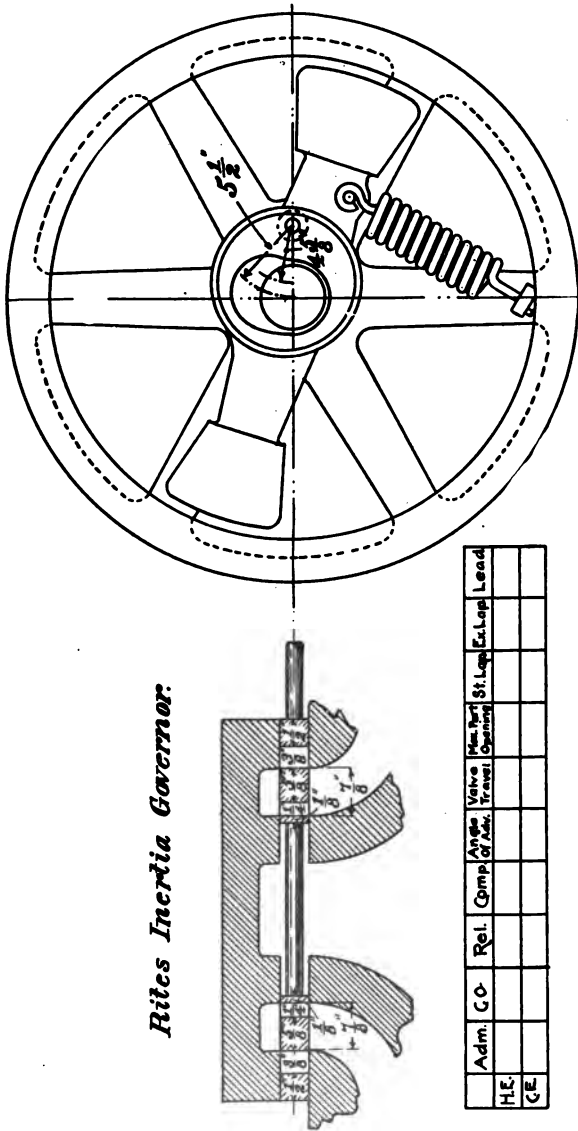


FIG. 171.

241. Assignment:—

Make analysis of governor and Zeuner diagrams for three assigned cut-offs by either one of the two following methods:

- (1) Equal cut-offs (20% to 75%) Per Cent.
- (2) Head end cut-offs (15% to 80%) Per Cent.

MECHANISM OF THE CENTRIFUGAL GOVERNOR.

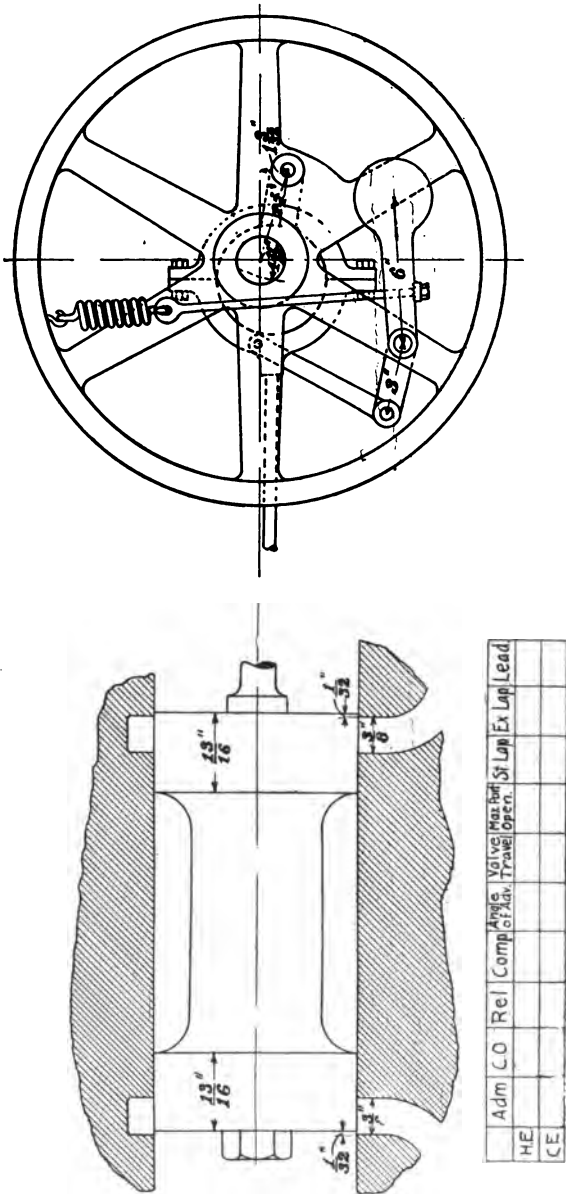


FIG. 172.

242. Assignment:—

Make analysis of governor and Zeuner diagrams for three assigned cut-offs by either one of the two following methods:

- (1) Equal cut-offs (20% to 75%) Per Cent.
- (2) Head end cut-offs (15% to 80%) Per Cent.

MECHANISM OF THE STRAIGHT LINE GOVERNOR.

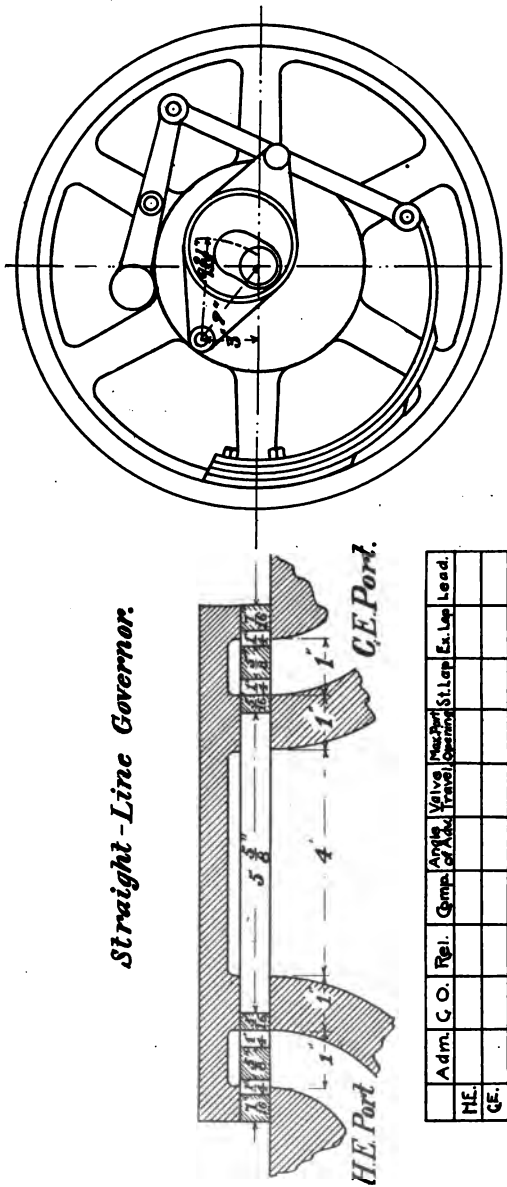


FIG. 173.

243. Assignment:—

Make analysis of governor and Zeuner diagrams for three assigned cut-offs by either one of the two following methods:

- (1) Equal cut-offs (20% to 75%) Per Cent.
- (2) Head end cut-offs (15% to 80%) Per Cent.

MECHANISM OF THE STEPHENSON LINK.

(See "Anchored" Valve Motions)

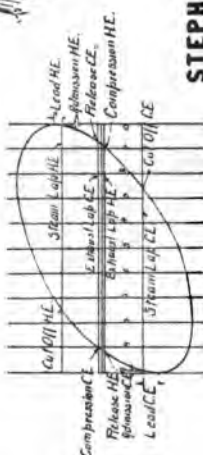


Table of Events

	Lead	C/O	Release	Compress
HE	15%	75%	15%	15%
CE	15%	75%	15%	15%

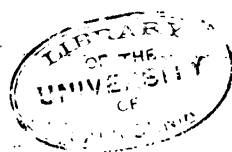
Link Diagram Shows of 80% Stroke

STEPHENSONS LINK

VALVE ELLIPSE

ANALYSIS.

244. Assignment:—In this analysis, assign the cut-off (20 to 80 per cent.) or the lead. Set the link to give this cut-off and draw in position of cut-off. Finally draw valve ellipse and fill in table of events



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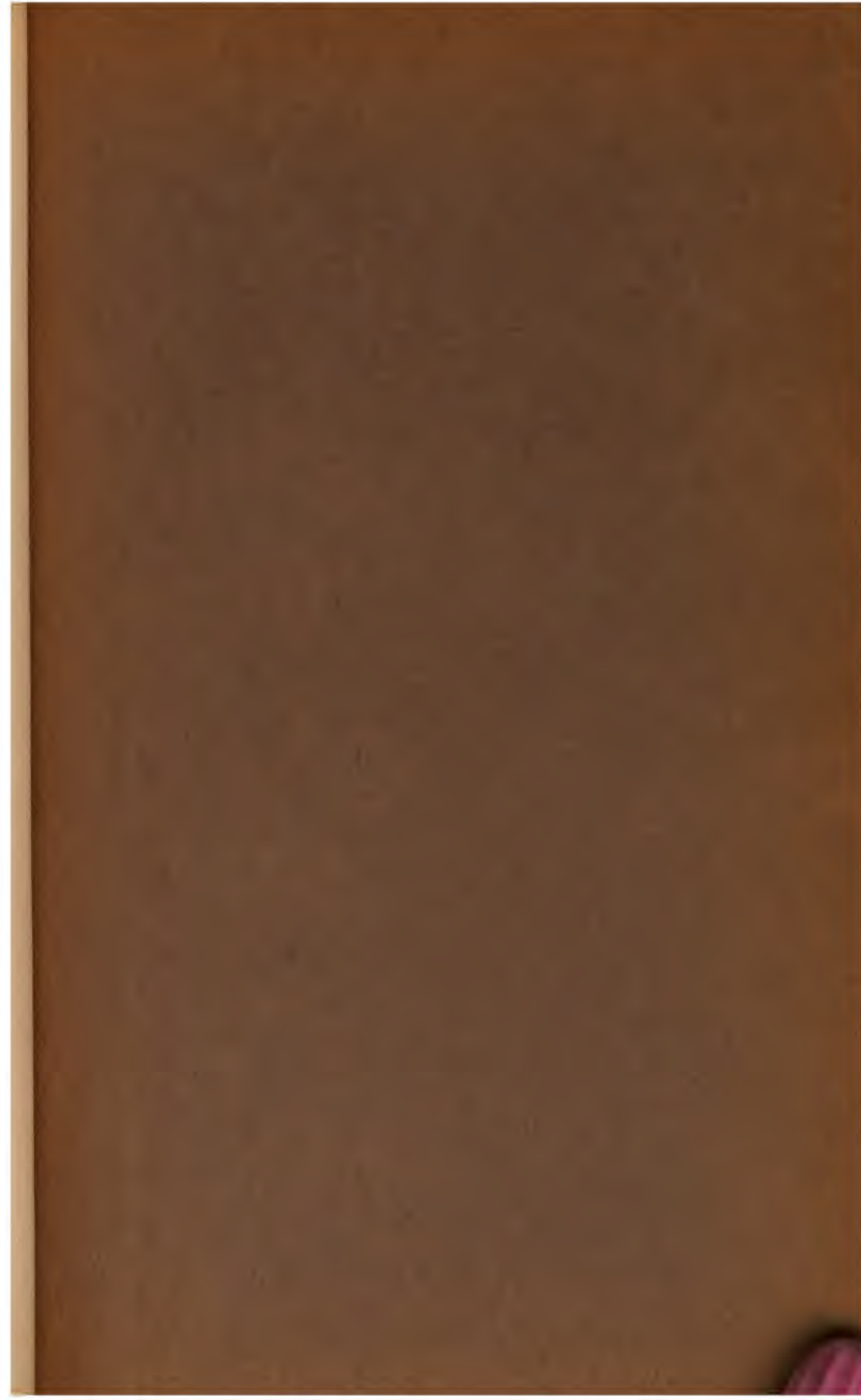
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